Title

Testing of High speed Internal Combustion engines

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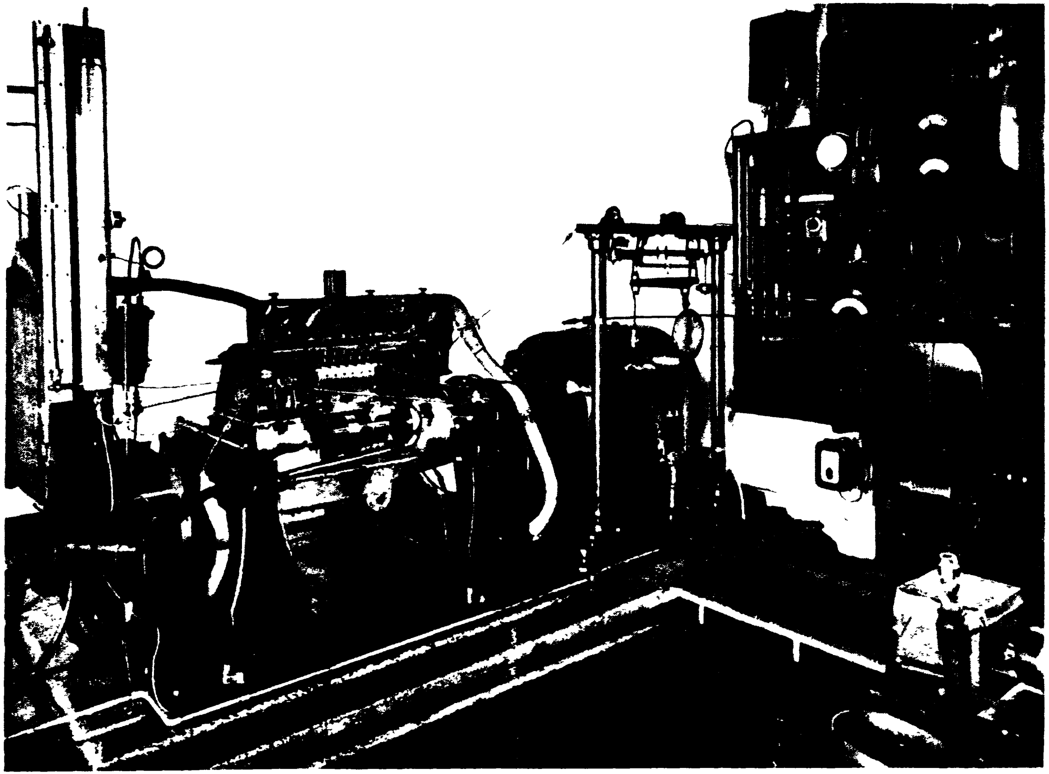

Showing the A.E.C. high-speed Diesel engine testing equipment. The fuel-measuring apparatus is shown on the left-hand side.

[*Frontispiece.*

PREFACE TO THIRD EDITION

DURING the interval that has elapsed since the publication of the last edition there have been considerable developments in engine testing methods and appliances; these have rendered necessary a fairly complete revision of the text matter and illustrations.

Certain sections of the previous edition have been re-written and expanded, whilst new chapters on Cathode Ray Indicators and Testing of High Speed Compression-Ignition Engines have been added.

In the earlier portion of this book, the matter relating to basic principles, calculations and formulæ, involved in engine test and research, has been revised and extended, appreciably.

Other sections that have been more fully revised are those relating to Dynamometers, Instruments and Aircraft Engine Testing Methods and American Test Equipment.

Unfortunately, owing to present War censorship conditions, it has not been possible to include certain more recent items relating to aircraft and automobile engine tests, but it is believed that the present volume is otherwise up to date.

Advantage has been taken of the opportunity afforded by the necessity of reprinting this third edition to include an extra Appendix (No. VI) containing an account of the regenerative dynamometers employed for aircraft engine testing purposes, that are now being widely adopted.

A. W. JUDGE.

FARNHAM, SURREY,
1944.

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Chapter I

GENERAL PRINCIPLES

Introductory.—The comparatively high standard of efficiency and reliability of the modern high speed internal combustion engine is due very largely to the application of test and research information and results; it can be stated that the only satisfactory method of developing present-day engines is along the same lines. In the early days of the petrol engine, the designer worked in an atmosphere of doubt and uncertainty, but in the course of time he gradually acquired information and design data, as a result of constantly testing, adjusting, and making changes in the existing designs. The application of the test data enabled him to improve radically his products, to formulate theories in order to explain the observed phenomena, and to predict, as far as possible, the effect of certain changes in the design. Very often, also, whilst searching for an explanation of one observed effect, he stumbled upon another and much more important result; in this connection the same possibility even to-day exists.

Although a considerable amount of useful information has been obtained from time to time, during the progress of petrol engine design, from the results of tests and researches, yet, on the other hand, a certain number of these tests have proved valueless on account of ignorance on the part of the experimenter of the real principles underlying this branch of work.

One has seen the results of many months of arduous tests upon particular engines completely vitiated by neglect on the part of the initiator to take account of a certain variable which influenced the results of his other observations. Thus in one case, the results of a number of tests upon a certain design of automobile engine which were published appeared very detrimental to the engine, but the subsequent discussion on the published paper revealed the fact that the ignition setting was incorrect, there was a serious wire-drawing effect at the throttle of the carburettor, and that the engine was over-cooled.

Therefore, it is necessary for the experimenter to make himself familiar with the known basic principles underlying the testing of engines, and to acquaint himself with the various factors concerned, before undertaking any serious work.

Classification of Tests.—The testing of high speed internal combustion engines usually falls into one of three categories, as follows: (1) Routine and Acceptance Tests; (2) Tests of New

Types, to ascertain the effects of certain design changes; and (3) Analytical or Research Tests.

The former type of test is that generally carried out at production works, in order to ascertain whether each engine gives about the correct output, and to test its reliability. Such tests constitute the simplest of those which we shall discuss, but nevertheless they are of equal importance to the others.

Although modern manufacturing methods, involving as they do a thorough system of inspection, gauging, and testing of the component parts, enable a high percentage of satisfactory engines to be obtained, yet it is only by actual bench and road tests that the final defects and performance are ascertained.

The second class of test is a more thorough one, and involves generally the use of additional apparatus and methods. It includes the experimental designs of manufacturers' engines which are intended for the market at a future period. The modern tendency towards higher and higher efficiency and output from a given size of engine, and the existence of competition among rival firms, compels the manufacturer constantly to improve and redesign his products. As a result of his customers' and his own experiences during the season, certain shortcomings or defects are revealed; very often his own product does not compare quite favourably with that of another firm, on the road or in competitions. Often, also, during the period of running of his present models, a new development will be announced from another source. Thus, as the result of research, an authority may publish a statement that a certain design of valve, or combustion head gives far better results than the existing designs, and the progressive manufacturer, ever seeking for improvements, will "test out" the commercial possibilities of this development. It will thus become clear that the second type of test is a most important one; in it is wrapped up the whole future of petrol engine design, as far as the commercial products are concerned.

The third type of test comes into the domain of scientific research. It is undertaken, as a rule, to probe out the reasons for complex actions known to occur, to test the validity of explanatory theories, which may eventually have great influence upon design, and to analyse minutely the behaviour of existing and new types with a view to indicating the respects in which they can be improved.

The only criterion whereby the performances and possibilities of entirely new types of engine can accurately be gauged is that of research results. Thus a new type of engine will be set up on a test-bed, and its speed, torque, fuel, and oil consumptions, brake and indicated power, volumetric and thermal efficiencies, and heat balance, etc., ascertained. From the results, a comparison is made with those of existing, and ideal, standard engines, and a definite

opinion can be formed as to its possibilities. Very frequently, also, the processes of the investigation will reveal the directions in which beneficial changes can be made.

In the following pages the basic principles involved in testing will be touched upon, where it is considered necessary, in order to enable the reader to form a broad idea of the test and research requirements, before plunging into details and descriptions. We shall confine our remarks, largely, to the petrol and Diesel types of high speed internal combustion engine, although most of the methods and results will be found directly applicable to other types of engines.

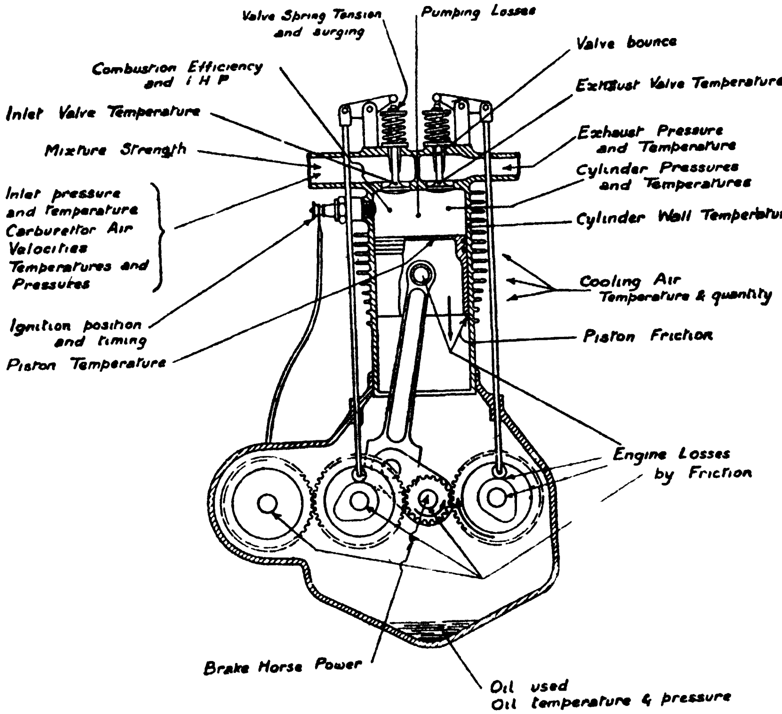


FIG. 1.

Factors Involved in Petrol Engine Tests.—A brief survey of the different quantities involved in the operation and testing of petrol engines is necessary as a preliminary, or introduction, to the later sections. Moreover, as we have previously mentioned, a sound knowledge of these factors is essential, if the experimental results are to be of real value. An elementary notion of the different variables and quantities with which the experimenter is concerned may be obtained from the diagram shown in Fig. 1. This represents, in outline, a typical air-cooled petrol engine, with the various sources of measurement, and the quantities to be measured indicated

thereon. At first sight, the numerous items shown may appear to be somewhat formidable, and the beginner or non-technical reader may have certain qualms concerning his ability to carry out the tests shown; we would mention, however, that it is unnecessary for ordinary purposes to undertake all of these, and that the majority of the items shown belong to the realm of the scientific experimenter. For most practical purposes only three or four of the simpler types of test need be carried out, and, with a certain amount of training and intuition, sufficiently accurate results are obtainable, without much technical knowledge.

Referring to Fig. 1, the principal variable quantities concerned may be enumerated as follows:—

1. The speed of the engine crankshaft.
2. The proportion of air to fuel, i.e. the mixture strength.
3. The quantity of mixture admitted, or throttle opening.
4. The moment of occurrence of the spark.
5. The horse-power developed in the cylinder, i.e. the indicated horse-power, or I.H.P.
6. The horse-power developed at the crankshaft, i.e. the brake horse-power, or B.H.P.
7. The pressure of the gases or mixture in the cylinder.
8. The temperature of the gases or mixture in the cylinder.
9. The temperature of the cooling water, or air.
10. The inlet and exhaust pressures.
11. The inlet and exhaust temperatures.
12. The quantity of lubricating oil used, and its pressure (if pressure-fed).

This list may, of course, be elaborated, in the case of special tests and research work, involving other physical measurements. Thus it is sometimes necessary to measure the temperatures of the cylinder wall, the valves and the piston, the heating of the charge from the carburettor (where exhaust or a hot-water jacketing is employed), the heat lost by direct conduction through the cylinder walls, and also by radiation, torsional vibration frequencies and amplitudes, vibrational effects, valve spring characteristics, and other quantities. Reference will be made, later, to methods of carrying out certain special tests.

Variables to Consider.—The particular quantities with which the ordinary test assistant is concerned are those given in (1), (2), (3), (4), (9), and (12), all of which are variables. It is imperative, therefore, when making special tests to ascertain which of the variables to maintain "constant" for the period of the test, and which to "vary." Obviously there is a very large number of combinations of these quantities, each of which will give a different result. Thus one may endeavour to keep any five of these items constant

during a test and vary the sixth one throughout its range. Since, however, each of the other five may have a number of constant values, the number of possible experiments becomes considerable.

Fortunately, the question is not quite so complex as it would at first appear, since we have at our disposal the accumulated results of experience, and a general knowledge of the influence upon the performance of each of the above test factors. Thus we are able to assign the values of certain of these factors, so as to obtain the best results, from the point of view of power and fuel-economy say, and having obtained the best settings for these, then to vary the others as we require.

As an example, let us suppose that it is required to obtain the greatest output from a given engine, without, however, making any radical alterations to its design. Experience shows that first of all the mixture strength must be rather richer than for complete combustion, and that the spark should occur as early as possible. Thus we make a tentative "advance" setting of the ignition timing lever first, and proceed to run the engine for a time partly throttled, until everything, including the oil, has settled down to its normal working temperature.

Next, assuming that there is some means of absorbing and of measuring the power, namely, an absorption dynamometer, the throttle is opened full, and the mixture strength varied progressively from a weak (in fuel) state to the rich condition which the brake-horse-power measurements indicates to be the best. Having settled the mixture quality and quantity question, the carburettor settings left at these positions, and the position of the spark timing lever varied until the best power result is obtained.

Finally, with the above best settings, the temperature of the cooling water is raised or lowered until once again the best power output is obtained, and maintained.

In a similar manner the effects of oil pressure and quantity can be ascertained, and the best values decided upon.

The speed at which the greatest power is obtained can be found experimentally by varying the load on the brake, and taking a series of brake-load and speed readings, until the product of the two becomes a maximum. This maximum does not, in general, occur at the highest attainable speed of the engine but at a lower value, as will be observed from the typical horse-power speed curves given later.

Sufficient emphasis, it is believed, has now been given to the importance of a knowledge of the variables to take into account, and of controlling those which may influence the measurements made upon the others. The systematic method previously outlined will be found to save much valuable time, and also to enhance the value of the results obtained.

Influence of Certain Factors.—In order that the experimenter may be saved the trouble of finding out for himself the influence of some of the factors mentioned, it is proposed to consider briefly a few of the more important experimental results of noted authorities, in so far as they bear upon the present subject. It is not proposed, however, to enter into details concerning the methods of test, the underlying theories, or the subsidiary facts; these, we feel, should be studied from the original papers and treatises mentioned in the Bibliography at the end of this book.

(*a*) **Effect of Mixture Strength upon Power and Economy.**—The results of a number of accurate tests made by Watson, Hopkinson, Ricardo, and others, show conclusively that the mixture strength (i.e. the ratio of air to fuel) which gives the greatest power output, other conditions remaining the same, is one lying between the mixture proportions giving the most complete combustion, and the richest-in-fuel mixture upon which the engine will run. In the case of high-grade petrol, an engine will work over a range of from about 8 to 20 parts of air to petrol by weight. Commencing with the latter weak mixture and progressively enriching the mixture it is found that at about 14·7 the combustion of the fuel is correct, that is to say, all of the petrol is combusted with all of the oxygen of the air. The mixture which gives the greatest mean effective pressure (or power, in this case) is one about 20 per cent richer than this, that is, of value about 11·8, and for this mixture strength about 4 per cent. higher m.e.p. is obtained.

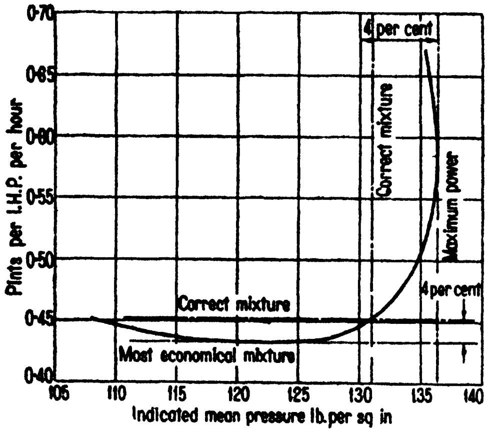


FIG. 2.—Showing the variation of the m.e.p. with mixture strength.

Fig. 2 illustrates the manner in which the m.e.p. varies with the mixture strength.

The mixture which yields the best fuel economy for a given power output is not the chemically correct one, nor is it the maximum power one, but a rather weaker mixture than the former, viz. one of about 13 per cent. more air, i.e. a mixture of about 17 parts of air to 1 part petrol.

(*b*) **Effect of Nature of Fuel.**—It is very important in connection with tests, especially comparative ones, to know the

composition and heating value of the fuel used. A considerable variation in the performance and output of an engine may be obtained by using different fuels.

The composition of all liquid fuels used in high speed engines consists of the elements carbon (C), hydrogen (H), and occasionally oxygen (O), with traces of impurities such as sulphur in a few instances.

Table I gives the principal properties of the more common fuels in use, namely, petrols, benzole, alcohol, and paraffin.

TABLE I

Properties of Various Fuels

Name of Fuel	Chemical Formula	Approx. Spec. Grav.	Composition per cent. (wt.)			Calorific Value in B.T.Us. per lb.
			Carbon	Hydrogen	Oxygen	
Alcohol, 100 per cent.	C_2H_6O	·795	52·2	13·0	34·8	11,600
Methyl alcohol	CH_4O	·829	37·5	12·5	50·0	9,500
Methylated spirit	Ethyl alcohol, Wood spirit, Naphtha (Mixture)	·821	—	—	—	11,000
Benzole		·87-·89	92·3	7·7	—	18,000-18,500
Petrol ·680 density †	,,	·680	84	16	—	19,200 *
·720 ,,	,,	·720	—	—	—	18,700 *
·760 ,,	,,	·760	—	—	—	18,250 *
Paraffin	—	·79-·81	85	15	—	18,900
Hexane	C_6H_{14}	·861 ‡	83·7	16·3	—	20,000
Heptane	C_7H_{16}	·867 ‡	84	16	—	20,760
Pentane	C_5H_{12}	·640 ‡	82·9	17·1	—	18,410
Toluol	CH_8	·861	91·3	8·7	—	18,300
Xylol	C_8H_{10}	·867	90·5	9·5	—	18,460
Acetylene	C_2H_2	gas	92·3	7·7	—	21,600

* Higher heating values.
† Consists of 80 Hexane, 18 Heptane, 2 Pentane.
‡ At 5° C.

The *calorific* or *heating* value of a fuel is the amount of heat evolved during the combustion of unit weight of the fuel with air or oxygen. It is usual to express the calorific value in terms of the number of British Thermal Units (B.T.Us.) given out by the complete combustion of 1 lb. of the fuel. The combustion of a hydrogen containing fuel results in water (H_2O) being formed as a product, and this water is in the form of steam. The extra amount of heat obtained when the steam condenses is equal to the latent heat of the quantity of water present (i.e. 966 B.T.Us. per lb. condensed).

The " higher heating value " of a fuel includes this extra amount of heat, whereas the ordinary or lower heating value disregards it.

The heating value of any fuel whose composition is known can readily be calculated from the relation

$$\text{Calorific value} = 14{,}500\ C + 52{,}500\left(H - \frac{O}{8}\right),$$

where C, H, and O are the proportional weights (expressed as decimals) of carbon, hydrogen, and oxygen present.

It will be observed that the higher the hydrogen content, the higher is the calorific value, and that the effect of oxygen in the fuel is to reduce the heating value.

In petrol engine calculations it is usual to assume the lower heating values, since the water formed during combustion does not condense in the cylinder.

The quantity of air theoretically necessary for complete combustion can be calculated from the chemical formula of the fuel.

Thus, if we assume high-grade petrol to be nearly pure hexane C_6H_{14}, the chemical equation for combustion is

$$C_6H_{14} + \left(6 + \frac{14}{4}\right)O_2 = 6CO_2 + \frac{14}{2} \cdot H_2O.$$

Molecular weight : 86 304 264 126

From which it follows that 1 lb. of petrol requires $\frac{304}{86} = 3{\cdot}53$ lb. of oxygen for complete combustion. Since, also, atmospheric air consists of 76·8 per cent. nitrogen and 23·2 per cent. oxygen, the equivalent weight of air to 3·53 lb. of oxygen is $\frac{100}{23{\cdot}2} \times 3{\cdot}53 = 15{\cdot}21$ lb. Thus 1 lb. of petrol requires 15·21 lb. of air for combustion.

Anti-Knock Fuels.—From the above brief facts the importance of specifying the exact nature of the fuel employed in the tests will be realized. In this connection it may be mentioned that every fuel has a limiting value of compression pressure when employed in a petrol-type engine, and in the case of a low-limit fuel the power developed by the engine is limited by this fact. Any attempt to run this engine above the limiting compression results in detonation.

However, by mixing a low-limit fuel with a high-limit one, it is at once possible to raise the value of the compression, and thus to obtain a greater power output, and, generally speaking, a better fuel economy.

Similarly, by adding what are known as anti-knock constituents or "stabilizers," such as tetra-ethyl-lead, ethyl iodide, aniline (derived from coal-tar) and xylidine in small percentages to ordinary petrol, the limiting-compression (which is about 140 lb. per sq. in.) can be raised considerably and more power obtained without the tendency to " knock " or detonate.

The hydrocarbons used for petrol engine fuels include the *aromatics, naphthenes, paraffins* and *unsaturateds.*

The aromatics include benzene (C_6H_6), toluene (C_7H_8) and xylene (C_8H_{10}) with specific gravities of 0·884, 0·870, and 0·862 respectively, and boiling-points of 80°, 110°, and 84°-140° C. (at 15·5° C.), respectively. The corresponding calorific values are 17,260, 17,510, and 17,610 B.T.Us. per lb. respectively. These fuels have the highest H.U.C.R. or anti-knock values of the constituents previously mentioned, but they have the lowest calorific values.

The naphthenes, represented by the general chemical formula, C_nH_{2n}, of which cyclo-pentane (C_5H_{10}) and cyclo-hexane (C_6H_{12}) are examples, have specific gravities ranging from 0·75 to 0·78; boiling-points from 50° to 95° C., and calorific values of 18,660 to 18,790 B.T.Us. per lb.; the latter, it will be seen, are appreciably higher than for the aromatics.

The paraffins, which include a larger number of constituent fuels of the general formula C_nH_{2n+2}, have, in general, the highest

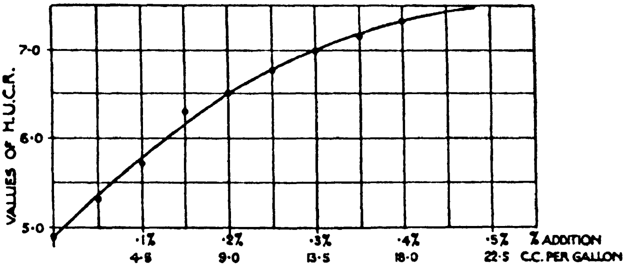


FIG. 3.—Effect of tetra-ethyl lead on H.U.C.R.

calorific values, the lowest specific gravities and—in the case of the members known as heptane, octane, iso-octane, nonane and decane—the highest boiling-points.

The unsaturateds have the same general formula, namely, C_nH_{2n} as the naphthenes; specific gravities ranging from 0·65 to 0·75; boiling-points from 40° to 178° C., and calorific values from about 18,700 to 19,700 B.T.Us. per lb.

Whilst the unsaturateds have higher knock ratings than the paraffins, they tend to become oxidized by the atmosphere, with subsequent gum formation; to counteract this tendency special inhibitors are employed.

Tetra-Ethyl Lead.—The so-called "leaded" petrols used in high compression petrol engines are petrols containing tetra-ethyl lead fluid, a typical one of which, used for aircraft fuels, is that known as "I.T. ethyl fluid." It contains about 61·4 per cent. (by weight) of lead tetra-ethide, 35·7 per cent. of ethyl dibromide, 0·17 per cent.

of blue colouring matter, and a small proportion of kerosene and impurities.

This fluid has a specific gravity of 1·755 at 20° C. and actually contains 65·5 per cent. of tetra-ethyl lead by volume.

The ethyl dibromide is a colourless liquid, about twice as heavy as petrol, and is included to prevent the lead from burning to lead

TABLE II

Combustion Properties of Fuels (Ricardo)

Name of Fuel	Spec. Grav.	Heat of Combustion in Ft.-Lb. per Cu. Ft.	Air to Fuel by Wt. for Complete Combustion	Max.* Compression Ratio which has been used satisfactorily	Corresponding* Compression Pressure Lb. per Sq. In.	Lowest Fuel Consumpt. Lb. per I.H.P. Hour	Thermal Efficiency at Max. Comp. Per Cent.	Max. Ind. m.e.p. at Max. Comp.
Pure petrol (aromatic free)	·718	46·08	15·05	4·85	105·5	·422	31·4	138·1
"D" petrol .	·760	46·18	14·6	5·35	121·5	·407	33·1	142·9
Paraffin .	·813	46·14	15·0	4·2	86·0	·581	22·9	—
Pentane .	·624	46·25	15·25	5·85	138·5	—	—	—
Hexane (80 per cent.)	·685	46·0	15·2	5·1	113·5	·405	32·4	141·2
Benzene (pure)	·884	46·9	13·2	6·9†	179	·392	37·2	156·0
Toluene (pure)	·870	46·9	13·4	>7·0	>183·0	·385	37·5	156·3
Xylene (pure)	·862	46·7	13·6	>7·0	>183·0	·381	37·3	156·1
Cyclohexane (93 per cent. pure) . .	·786	46·08	14·7	5·9†	140·5	·385	34·9	148·0
Ethyl alcohol (98·5 per cent.) .	·796	44·5	8·9	>7·5	>204·0	·532	32·4	165·5
Methyl alcohol .	·826	45·5 (app.)	6·5	5·2†	116·5	·725	32·7	153·9
Methylated spirit .	·821	44·0	8·0 (app.)	6·5†	>163·5	·625	32·5	165·0
Ether (50 per cent. in petrol) . .	·727	46·4	13·0	3·9	77·0	—	—	132·5
Carbon disulp. (50 per cent.)	·994	40·2	10·8	5·15	115·0	—	—	136·3
Heavy aromatics . .	·835	46·66	13·8	6·5	163·5	·447	31·5	—

* As tested in Ricardo's Variable Compression Engine.
† Pre-ignition occurred before audible detonation.

oxide and depositing itself in the combustion chamber; instead, it mostly passes out through the exhaust in the form of a vapour.

Another ingredient included in the above analysis is halowax oil, the object of which is to prevent the exhaust valve stems from becoming dry with consequent risk of seizure in their guides.

An important property of tetra-ethyl lead, previously referred to,

is the relatively small amount that is needed to raise the H.U.C.R. of the petrol appreciably. Thus, less than one-half of 1 per cent. of tetra-ethyl lead was found to raise the H.U.C.R. by as much as 50 per cent. in some tests carried out on the Ricardo Variable Compression Engine (Fig. 3).

It may be useful to include here a table showing the principal combustion properties of the available fuels which are applicable to high speed internal combustion engines, as these values will form a useful guide for comparative purposes. (Table II.)

Maximum Heat Energy in Charge.—The amount of heat energy available by combustion of the fuel and air depends upon the amount of oxygen present in the air charge.

Thus if V = working volume of cylinder in cu. in. and E_V = volumetric efficiency, then $E_V V$ = volume of fresh mixture drawn into the cylinder per suction stroke. Denoting the calorific value of the fuel by C_F and the proportions of air to fuel, by volume, by x we have

$$\text{Heat Energy of Charge} = E_V V \frac{C_F \times J}{x \times 1{,}728 \times 12{\cdot}4} \text{ ft. lbs.}$$

where $J = 778$ ft. lbs.

Taking $C_F = 19{,}300$ B.T.U. per lb. and $x = 15{\cdot}5$, this expression reduces down to the following :—

$$\text{Heat Energy of Charge} = E_V \,.\, V \times 45 \text{ ft. lb.}$$

In the ideal case where $E_V = 1$ the energy which is theoretically available works out at 45 ft. lb. per cu. in.

This is equivalent to about 100 B.T.Us. per cubic foot of mixture.

In the case of petrol of calorific value 18,700 B.T.Us. per lb. it can be shown that the maximum heat energy available in the charge is given by the following expression :—

$$\text{Heat Energy in Charge} = 99{\cdot}19 E_V \,.\, V, \text{ B.T.Us.}$$

In the ideal case, when the volumetric efficiency $E_V = 1$ we get an equivalent heat energy of 99·19 B.T.Us. per cu. ft. of mixture.

Mean Effective Pressures.—It is possible to estimate the ideal and also the actual indicated M.E.P. values from the data given in the preceding paragraph.

Assuming that the engine is using fuel of calorific value equal to 19,300 B.T.Us. per lb.—a value rather higher than for lighter petrols—we have

Energy liberated by complete combustion of the fuel = 45 ft. lb. per cu. in.

Assuming the thermal and volumetric efficiencies are each at their ideal value of 100 per cent., we have

$$\text{Theoretical I.M.E.P.} = 45 \times 12 = 540 \text{ lbs. per sq. in.}$$

Then, if the thermal efficiency be denoted by E_T and the volumetric efficiency by E_V,

$$\text{Actual I.M.E.P.} = E_T \times E_V \times 540 \text{ lbs. per sq. in.}$$

Thus if $E_T = 0{\cdot}30$ and $E_V = 0{\cdot}80$ the value of the mean pressure will be $0{\cdot}3 \times 0{\cdot}8 \times 540 = 129{\cdot}6$ lbs. per sq. in.

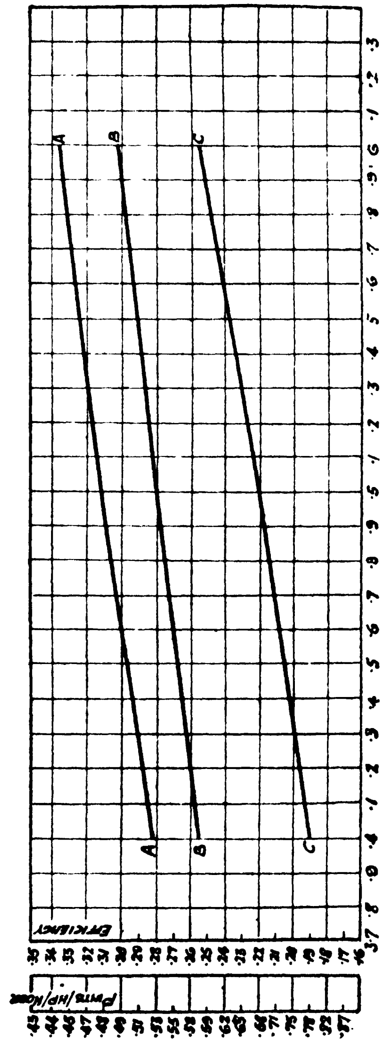


FIG. 4.—Showing relation between fuel consumption and compression ratio.

(*c*) **Effect of Compression.**—It can be shown from theoretical considerations that the explosion pressure, and also the mean effective pressure (m.e.p.) both increase progressively with the compression pressure, up to the limiting value of the latter, fixed by detonation.

For approximate purposes the following relations may be used, to afford data on the explosion and mean pressures :—

1. Explosion pressure = $3{\cdot}5$ to $4{\cdot}0$ (compression pressure).
2. M.E.P. = $0{\cdot}8$ to $1{\cdot}25$ (compression pressure).

Generally speaking, the lower efficiency engines give the lower value, and normal engines of modern design give a ratio of $1{\cdot}1$ to $1{\cdot}25$ in the latter case.

In the case of two cycle engines of the two and three port types, the explosion pressure is from $3{\cdot}0$ to $3{\cdot}7$ times the compression value, and the mean pressure from $0{\cdot}7$ to $0{\cdot}85$ times.

The fuel consumption per h.p. per hour also diminishes with increase in compression, so that from all points of view the use of the higher compressions is beneficial.

(*d*) **Thermal Efficiency.**—The basis upon which the performances of all internal combustion engines are compared is one which considers the amount of useful work obtained from the engine in relation to the heat energy supplied by the combustion of the fuel.

$$\text{The thermal efficiency} = \frac{\text{useful work obtained}}{\text{heat supplied by the fuel}}.$$

This ratio enables comparisons to be made between all types of internal combustion engines on a rational basis. The useful work obtained is usually taken as the "indicated" work, that is, the work calculated from the measured power, or Indicated Horse-Power (I.H.P.) in the cylinder itself, and the thermal efficiency thus reckoned is known as the Indicated Thermal Efficiency. If the Brake Horse-Power (B.H.P.) is used, then the Brake Thermal Efficiency is the one stated, and is, therefore, lower in value than

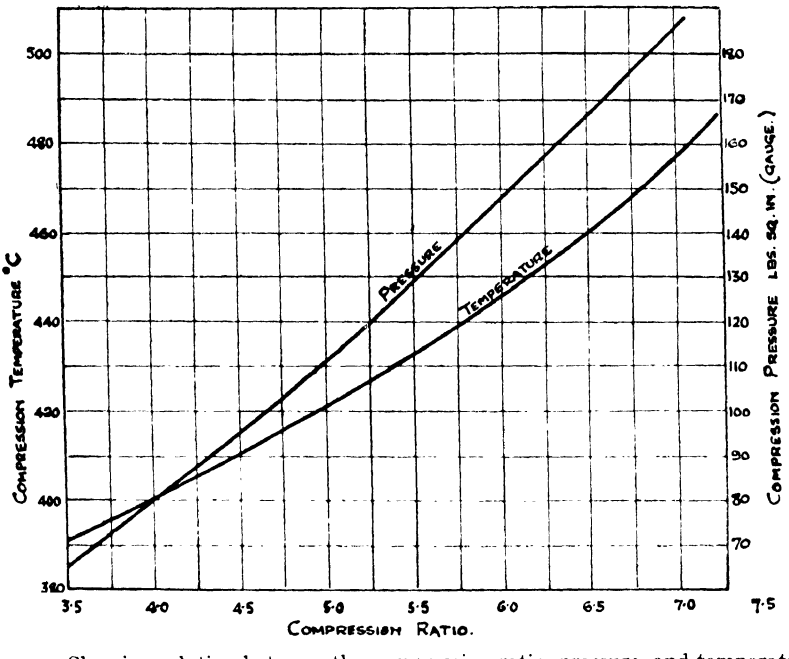


FIG. 5.—Showing relation between the compression ratio, pressure, and temperature.

the former efficiency. The heat supplied by the fuel is obtainable if we know the calorific value of the fuel and the fuel consumption of the engine ; both of these are measurable in engine tests.

Denoting the measured I.H.P. by the letters I.H.P., and the fuel consumption by W (lb. per I.H.P. hour), then, if C denotes the calorific value of the fuel (B.T.Us. per lb.), we have

$$\text{Thermal efficiency} = \frac{\text{I.H.P.} \times 33{,}000 \times 60}{W \times \text{I.H.P.} \times C \times 778}$$

$$= \frac{2540}{WC}.$$

Example.—An automobile engine having a mechanical efficiency of 90 per cent. gives 50 h.p. on the brake test, and uses 0·50 lb. of petrol of calorific value 18,000 B.T.Us. per lb. per B.H.P. hour.

The fuel consumption per hour will evidently be $50 \times \cdot 50 = 25$ lb.

The I.H.P. is given by $\frac{\text{B.H.P.}}{\cdot 90} = \frac{50}{\cdot 90} = 55{\cdot}5.$

The fuel consumption per I.H.P. hour is therefore $\frac{25}{55{\cdot}5}$.

Hence the indicated thermal efficiency

$$= \frac{2540}{\frac{25}{55{\cdot}5} \times 18{,}000} = \cdot 314, \text{ i.e. } 31{\cdot}4 \text{ per cent.}$$

Further, the brake thermal efficiency $= \cdot 90 \times 31{\cdot}4 = 28{\cdot}26$ per cent. Values of the thermal efficiency for various fuels and compressions will be found in Table II.

The results of numerous scientific investigations have established the following facts concerning the indicated thermal efficiency (T.E.) of the internal combustion engine :—

1. The T.E. depends upon the compression ratio. From both theoretical considerations of air cycle efficiencies and practical test results it has been shown that under normal running conditions, i.e. without detonation, the thermal efficiency of the high speed petrol engine can be expressed by the following relation :—

$$\text{Thermal efficiency} = 1 - \left(\frac{1}{r}\right)^n,$$

where r = compression ratio and n = a constant of value depending upon the mixture ratio and certain other minor engine design factors.

In this connection it is of interest to consider the results of an investigation made by Tizard and Pye [1] into the process of combustion and the limits of possible thermal efficiency in the case of hydrocarbon fuels.

It was found that the gain in efficiency with increase in compression ratio was greater than that predicted by the air cycle formula. Further, when the effects of dissociation and increase in specific heat with temperature rise were taken into account, the maximum temperature obtained with economical mixture strengths was practically the same for all hydrocarbon fuels usable in petrol engines ; with alcohol, however, it was rather lower.

The theoretical thermal efficiency for the most economical mixture,[2] assuming no loss of heat during the working stroke, but

[1] "The Character of Various Fuels for Internal Combustion Engines," Tizard and Pye, *The Automobile Engineer*, February, March and April, 1921.

[2] 20 per cent. weak in fuel.

taking into account the nature of the working fluid, is given by the relation

$$\text{Theoretical thermal efficiency} = 1 - \left(\frac{1}{r}\right)^{0.295}$$

In the case of the mixture chemically correct for combustion of the fuel the relation becomes

$$\text{Theoretical thermal efficiency} = 1 - \left(\frac{1}{r}\right)^{0.258}$$

The results of the investigations in question are shown graphically in Fig. 6, together with a curve of observed indicated thermal efficiency values obtained by Ricardo from a variable compression engine running on benzene with the most economical mixture strength. The air cycle efficiency curve is also shown on the diagram. A general similarity in the shapes and trend of all the curves is noticeable.

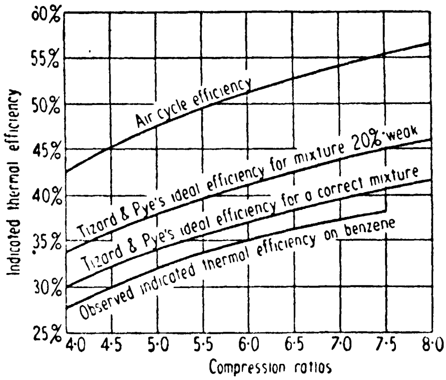


FIG. 6.—Thermal efficiency and compression ratio.

In regard to the quantitative effect of compression increase on efficiency the following table gives the results obtained from the Ricardo variable compression engine using benzene as a fuel :—

TABLE III

Compression Effect on Efficiency

Compression Ratio	Air Cycle Efficiency (1)	Observed Efficiency (2)	Ratio $\frac{(2)}{(1)}$
4 to 1	0·425	0·275	0·650
5 to 1	0·475	0·316	0·665
6 to 1	0·512	0·347	0·675
7 to 1	0·540	0·372	0·685
7·5 to 1	0·553	0·382	0·690

From the results given in the last column it is evident that the gain in efficiency is greater with increasing compression ratio than that predicted by the air cycle formula.

2. The T.E. increases with the engine speed up to the most economical speed. In the case of modern car engines this speed is

usually less than the maximum power output speed, by from 20 to 30 per cent.

3. The T.E. of an engine running light and at part loads is less than when running at nearly full load. The fuel-consumption of a partly throttled engine is, therefore, greater than when it is working at full throttle.

4. The T.E. of an engine operating at a given speed and throttle opening, and with other influencing factors remaining constant, depends upon the mixture strength.

In general the efficiency is a maximum for a mixture about 15 per cent. weaker, in petrol, than the one giving correct combustion conditions and is a minimum for weaker and also richer mixtures than this optimum value.

Fig. 7 illustrates the results of some tests made by H. Ricardo with different fuels on a single cylinder engine of 5 : 1 compression ratio running at 1,500 R.P.M. with a heat input to the carburettor of 0·0433 B.T.Us. per revolution.

The lower curves represent the indicated thermal efficiency values for the various mixture strengths shown at the tops of the ordinate lines. The upper curves give the corresponding indicated mean effective pressure (I.M.E.P.) values under similar conditions.

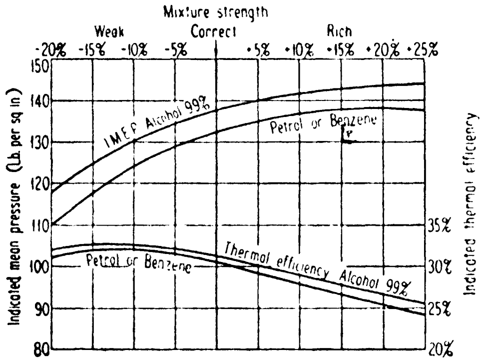


FIG. 7.—Thermal efficiency, I.M.E.P., and mixture strength.

It will be observed that for the fuels mentioned the maximum efficiency occurs for mixtures about 15 per cent. weak. On the other hand, the power output increases progressively as the mixture is enriched up to about 20 per cent. on the rich side. It follows, therefore, that the maximum output of an engine can only be obtained at the expense of greater fuel consumption per horse-power.

Another interesting feature shown in Fig. 7 is the greater efficiency and power output given by the air-alcohol mixtures over the whole range.

5. The T.E. depends upon the nature of the fuel employed in the engine. Thus, low octane fuels can only be employed with relatively low compression pressures—corresponding to low thermal efficiencies. On the other hand, the use of high octane fuels, namely, from 85 to 100 and above, enables much higher compression ratios to be employed without detonation occurring. For the same compression ratios, however, it has been shown by Tizard and Pye

that the various hydrocarbon fuels usable in petrol engines give about the same theoretical thermal efficiency values. The only exception is in the case of alcohol, which, on account of its high latent heat of evaporation (about two and a half times that of ordinary petrol), gives a slightly higher efficiency, by about 2 per cent. for a 5 : 1 compression ratio. Further, as this is a non-detonating fuel it can be used for compression ratios up to about 7·5 : 1, when it gives about 20 per cent. higher efficiency than the maximum value possible with ordinary petrol.

Most of the improvement in regard to the power output per unit volume of modern petrol engines—both unsupercharged and supercharged—has been due to the progressive increase in the octane number of the fuels made available. It has been shown that the

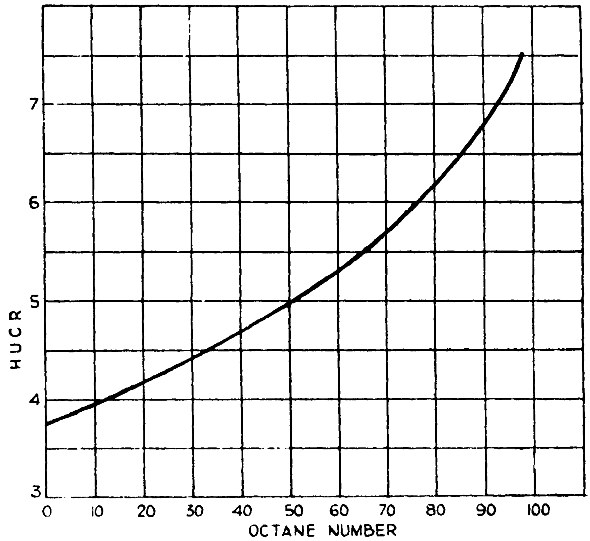


FIG. 8.—Relationship between H.U.C.R. and octane number of fuel.

highest useful compression ratio increases as the octane number is increased, as illustrated in Fig. 8, and since the thermal efficiency increases with the compression ratio it follows that it also increases with the octane number.

The results of fuel consumption tests upon an engine using fuels of different octane number confirm these conclusions for they show a progressive decrease in fuel consumption per H.P. hour with increase in octane value (Fig. 9).

The design of the combustion chamber has a pronounced effect upon the T.E. Generally speaking, the internal shapes giving the smallest amount of wall surface for a given volume are the most efficient. Thus, when the shape of the piston-head and the combustion chamber approximate to that of a sphere, the ideal

conditions are approached. For this reason the overhead valve engine with inclined valves and a dished piston is superior to the ordinary side-by-side valve engine in the matter of thermal efficiency. A special form of combustion head, due to Ricardo, enables relatively

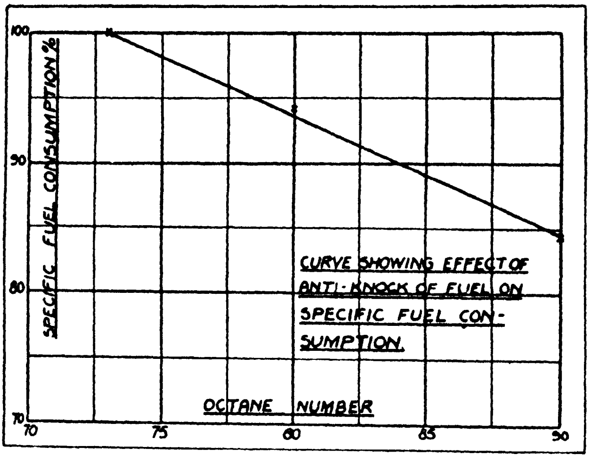


FIG. 9.—Fuel consumption and octane number.

high thermal efficiencies to be obtained from the side-by-side valve type of engine.

As a result of a careful study of the combustion process in petrol engines it has been shown [1] that there are different combustion zones in the cylinder head or combustion chamber. This is illustrated in the example of the modern cylinder head design shown in Fig. 10.

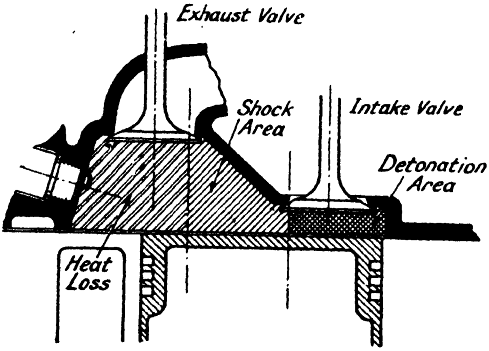


FIG. 10.—Combustion chamber zones.

It will be seen that the chamber is divided into three zones. The first zone is the ignition area which is likewise the heat loss area. In this area the metal is exposed to burning for the longest time, and, as pointed out by G. M. Rassweiler and Lloyd Withrow [2] of General Motors Research, is the area of highest temperature. This area should be protected against heat loss, and therefore the exhaust valve is placed there.

The second zone is the shock area, because during the time this area is burning the crankpin and piston are passing through

[1] *Aircraft Engines*, Vol. I, A. W. Judge (Chapman & Hall Ltd.).
[2] Trans. Soc. Automotive Engrs. U.S.A., Vol. 30, 1935.

top-dead-centre, and thus the structure is given the maximum effect of the pressure rise. It is desirable to reduce the pressure-rise rate in this area and hence a portion of the volume must be displaced from this point. If this is placed in the third area, which is the detonating zone, more volume-to-surface results than is satisfactory for detonation control. The relatively cooler inlet valve when placed in this area helps to absorb the heat of this super-compressed last portion of the charge to burn.

This is added to the last third, and, providing the depth of the section through this area is not over $\frac{5}{16}$ in., the detonation may be adequately controlled for compression ratios up to 6·5 : 1. In this manner it is possible to get a lower heat absorption at the first area, allowing lowering of the flame front in the shock area and yet providing additional heat in the detonating area.

It follows from these considerations that it is possible by employing a suitable design of combustion chamber to work with higher compression ratios, without risk of detonation when using a fuel of given octane value ; the thermal efficiency of an engine using this combustion chamber will therefore be higher than for the designs employed previously.

The T.E. depends upon the actual size of the engine, i.e. upon the cylinder bore, other conditions remaining the same. The smaller engines are the less efficient in this respect.

Other factors having a minor influence upon the T.E. include the valve-timing arrangement, the materials of the piston and cylinder, and the nature of the walls, i.e. whether carbonized or not.

(*e*) **Temperature of the Cylinder Walls.**—In order to obtain the greatest T.E. and power output from an engine it is necessary that the temperature of the cylinder walls and combustion head should be maintained at a certain minimum value. If too cool, an engine loses both in power and efficiency.

The cooling arrangements should, therefore, be so designed that the rate of heat flow through the cylinder walls should be maintained fairly constant and at the correct value.

The results of tests made by Hopkinson, Watson, and others indicated that for the best results the amount of heat flowing through the walls to the cooling water, or in the case of air-cooled engines to the cooling air, should be as follows :—

TABLE IV

Heat Extracted by Cooling Arrangements in Modern Engines

Speed in R.P.M.	1000	1400	1800	2000	2400	2800	3000
Heat extracted expressed as a percentage of the total heat of the fuel .	17·0	16·4	16·1	16·0	15·95	15·9	15·85

In this respect it was shown by Hopkinson that although the I.H.P. developed does not vary quite so much with cylinder temperature, yet the engine losses due to wall friction, which constitute the greater part of the total losses, diminish progressively as the cooling medium's temperature is raised. Thus in the case of a 16 h.p. engine, maintained at 800 r.p.m., he found the frictional losses to be 4·0 h.p. for a mean cooling water temperature of 65° F., 2·6 h.p. when at 150° F., whilst when the water was just below boiling-point, namely, 212° F., the losses were only 2·0 h.p.

Dr. Watson, when testing a single-cylinder sleeve valve engine, found that the B.H.P. rose from 4·6 when the cooling water temperature was 70° F., to 5·95 when it was 180° F. The I.H.P. remained constant at about 8·1 throughout.

Aircraft Engine Cooling Losses.—The results of more recent researches upon aircraft engines [1] indicate that from 25 to 30 per cent. of the heat of combustion of the fuel is absorbed or dissipated by conduction through the metal of the cylinders and pistons and by radiation. The proportion utilized as B.H.P. varies between 23 and 27 per cent. of the total fuel energy according to the engine design, load, speed, compression ratio, valve-timing, etc.

Part of the "cooling losses" is transmitted by conduction through the pistons and connecting rods and from the cylinder block to the crankcase where it is lost to the lubricating oil and by radiation and convection to the outside air; it has been estimated that in this way about 10 per cent. of the cooling losses is accounted for. The remaining 15 per cent. of the heat is disposed of by the process of conduction through the cylinder head and walls to the cooling water—or direct to the cooling air stream in the case of air-cooled engines. It is generally agreed that the amount of heat disposed of by the cooling system represents about 50 to 60 per cent. of the B.H.P. equivalent heat energy.

The quantity of heat to be dealt with by the cylinder cooling system, whether air- or water-cooled, may be estimated, approximately, as follows :—

Let w = fuel used in lb. per B.H.P. per hour. C_F = calorific value of fuel in B.T.Us. per lb.

Then total heat energy of fuel = wC_F B.T.Us. per hour.

Assuming that 25 per cent. of the heat energy is recoverable as useful work on the piston, i.e. as B.H.P., and that 60 per cent. of this represents the heat to be dealt with by the cooling system, we have

Heat dealt with by cooling system

$$= \frac{0{\cdot}25 \times 0{\cdot}6wC_F}{60}$$

$$= \frac{wC_F}{400} \text{ B.T.Us. per B.H.P. per min.}$$

[1] *Ibid.* page 18, 1st reference.

Thus in the case of an engine using 0·5 lb. of petrol (of calorific value 18,600 B.T.Us. per lb.) per B.H.P. per hour—

Heat dealt with by cooling system = 23·25 B.T.Us. per B.H.P. per min.

In practice the value in question varies between about 25 and 30 for aircraft engines running on ordinary fuels, but with high octane fuels and high compression ratios, the proportion of the heat to be dealt with by the cooling system diminishes. Similarly, with compression-ignition engines, as the thermal efficiencies are appreciably higher than for petrol engines, a greater proportion of the fuel's heat energy is usefully employed as B.H.P., leaving a relatively smaller proportion to be disposed of by the cooling system and exhaust.

(*f*) **Volumetric Efficiency.**—This is an important factor in connection with the power output of engines, and one with which the experimenter should be familiar. The amount of fresh mixture induced or drawn into the cylinder of an engine is theoretically equal to the piston-swept volume (at the same pressure as the outside atmosphere). Owing, however, to the detrimental influence of various factors, the actual quantity of fresh charge induced is invariably less than the theoretical quantity, the ratio of the actual to the theoretical amount being known as the Volumetric Efficiency.

In modern practice, in connection with aircraft and automobile engines, the volumetric efficiencies attain maximum values of from 75 to 85 per cent. at normal speeds.

The volumetric efficiency of an engine may be obtained simply by measuring the volume of air consumed by the engine by one of the methods described hereafter. Certain precautions must be taken, however, and the temperature, pressure, and humidity of the air flowing to the carburettor noted. If the total amount of air at normal pressure and temperature as measured by an air-flow meter be V cubic feet per hour, and the total piston-swept volume of the cylinders be denoted by V_0, then the volumetric efficiency of the engine is given by the expression $\frac{30N \cdot V_0}{V}$ where N is the r.p.m. of the engine ; this expression is true for any number of cylinders.

Other methods of determining the volumetric efficiency depend upon measurements taken from the indicator diagram.

Probably one of the best methods is that of the light spring indicator diagram. If V_1 denote the stroke volume, and V_2 the volume intersected on the compression curve by the atmospheric line, then we have

$$\text{Volumetric efficiency (V.E.)} = \frac{V_2}{V_1} \times 100 \text{ per cent.}$$

The volumetric efficiency would be 100 per cent. if the pressure at the end of the suction stroke were atmospheric. It is not possible,

however, in normally aspirated engines to obtain terminal atmospheric pressure, so that the V.E. is always less than 100 per cent. In order to obtain a standard of comparison, it has been suggested that, instead of assuming a terminal pressure of 14·7 lb. per square inch, one of 14·3 lb. per square inch be employed. This is equivalent to reducing the ideal volumetric efficiency by 5 per cent.

The density of the charge at the end of the suction stroke is also a measure of the V.E. Thus, if the proposed standard inlet temperature of 57° C. = 330° C. absolute be used, this corresponds to a density of ·827, so that the V.E. is 82·7 per cent. for a terminal suction pressure of one atmosphere.

Using Ricardo's proposed terminal pressure of 14·3 lb. per square inch, the ideally attainable V.E. is $\cdot 95 \times 82{\cdot}7 = 80{\cdot}5$ per

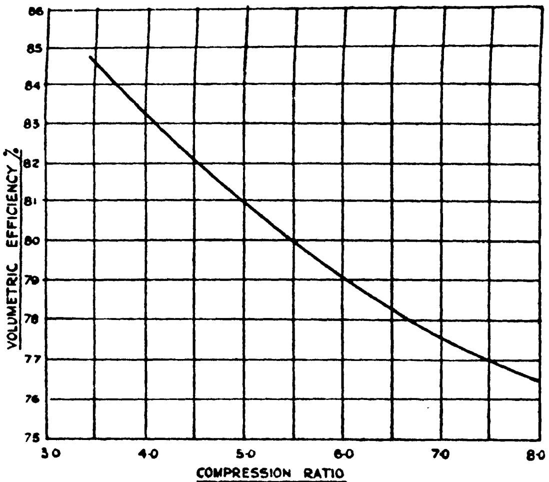


FIG. 11.—Effect of compression ratio on volumetric efficiency.

cent.; incidentally it may be mentioned that this corresponds to a gas velocity of 100 f.s. in a very short inlet pipe.

The following are the principal sources of loss of volumetric efficiency :—

1. Long and tortuous carburettor and inlet passages.

2. Insufficient inlet (and exhaust) valve area.

3. Premature heating of the charge during its passage into the cylinder, through the inlet port and valve passages. Heating reduces the charge density, and thus lowers the V.E.

4. Excessive friction of the mixture due to rough surfaces, and sudden changes in section of the carburettor and inlet pipe.

5. Incorrect valve timing. If the inlet valve opening period is not sufficiently long, or the points of opening and closing incorrect, the charge will be " wire-drawn," giving a drop in the V.E.

6. Insufficient valve lift, causing an increased velocity through the port, and consequent " wire-drawing."

7. Excessive back-pressure due to the exhaust gases. This may be due to insufficient exhaust valve area, or to bad silencer design.

For maximum values of the V.E., short inlet pipes of ample proportions, the minimum of bends and sectional changes, the minimum suction temperature, a good design of carburettor which does not wire-draw the charge are necessary.

The volumetric efficiency of a modern petrol engine *falls off with speed increase* after a certain maximum value has been attained at one particular speed, owing to the increase in resistance to the flow of the charge. *The volumetric efficiency* also *tends to fall off with increase in compression ratio* (Fig. 11). In this connection the results of an investigation made by C. C. Minter and W. J. Finn [1] with a variable compression engine are of interest. The engine was run with a constant inlet temperature of 100° F. and an evaporative cooling system giving a jacket temperature of 202° F. The engine was run at 600 r.p.m. on benzene with an air-fuel ratio of 10·85 and ignition advance of 12° It was found from theoretical considerations that the volumetric efficiency could be expressed in terms of the compression atio, as follows :—

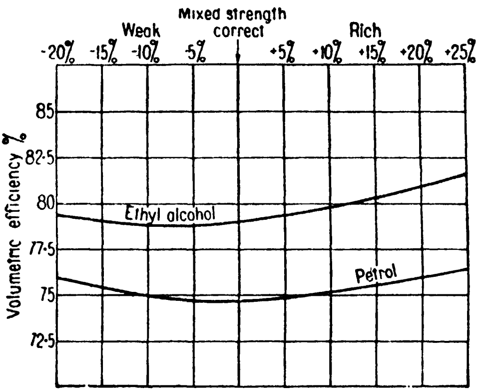


FIG. 12.—Mixture strength and volumetric efficiency.

$$E_V = A\sqrt{1 - \left(\frac{1}{r}\right)^{1\cdot4}} - B\left(\frac{r-1}{r}\right)$$

where A and B are constants and r is the compression ratio. The experimentally determined values of the constants were A = 177·4 and B = 108·2, the value of E_V being expressed as a percentage. The reason for this drop in efficiency is connected with the interchange of heat between the residual exhaust gases and the cylinder walls.

The volumetric efficiency is also dependent to some extent upon *the mixture strength;* the results of tests by Ricardo, with petrol and alcohol fuels (Fig. 12) illustrate this point. It will be observed

[1] Automotive Industries, July 27, 1935.

that the efficiency is a minimum for correct and slightly weak mixtures but is a maximum, in each case, for the richest mixtures. The reason for this is the effect of *the latent heat of evaporation of the fuel*, which results in a lowering of the charge temperature and therefore a corresponding increase in the charge density. So far as pure hydrocarbon fuels, such as the petrols, are concerned, the effect of latent heat is relatively a small one, but in the case of alcohol fuels the latent heat of evaporation is very much higher, so that the volumetric efficiency is also higher.

The volumetric efficiency is also dependent upon *the temperature of the inlet air*, its value being reduced as the air temperature increases.

The results of tests made upon a single-cylinder engine [1] having an electrically heated induction pipe for varying the temperature of

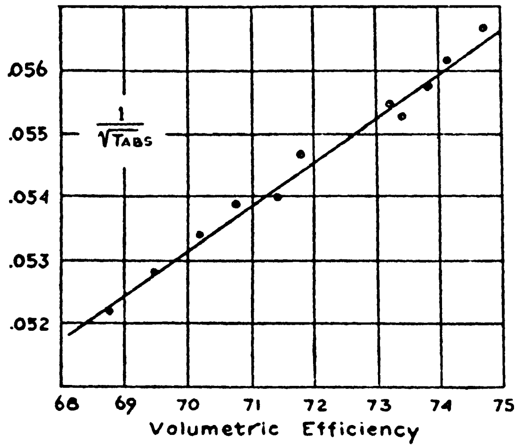


FIG. 13.—Inlet temperature and volumetric efficiency

the inlet air, in order to change the volumetric efficiency by this means alone, showed that instead of the volumetric efficiency varying inversely as the absolute temperature of the inlet air it *varied inversely as the square root* of the air temperature. Thus

$$E_v = \frac{K}{\sqrt{T}}$$

where K = a constant
T = absolute temperature.

These results, which are shown in Fig. 13 where values of $\frac{1}{\sqrt{T}}$ are plotted against the measured volumetric efficiencies, appear to be confirmed by those of Gibson and Gage. These experimenters

[1] *Ibid.* page 24.

found that as the inlet temperature was raised the volumetric efficiency did not diminish as rapidly as the air density, although the magnitude of the variations was not so great as those shown in Fig. 13.

The temperature of the cooling water has a small effect upon the volumetric efficiency, the latter increasing slightly with reduction in temperature of the water to the extent of about 1 per cent. for a fall of 30° C. from boiling-point (100° C.).

It may be of interest to conclude this section with a short reference to the sources of loss of V.E. in a specific instance representing

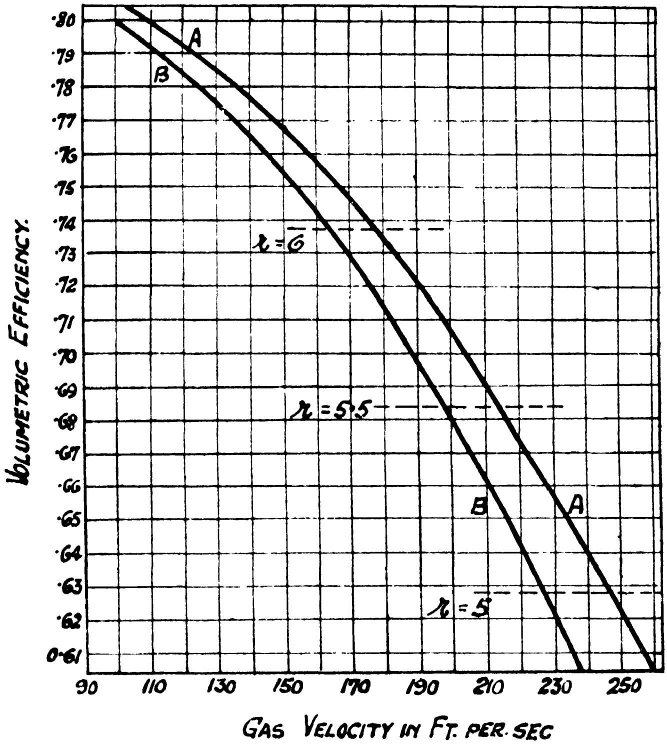


FIG. 14.—Showing relation between volumetric efficiency and gas velocity.

the practical idea attainable value, viz. 80·5 per cent. Fig. 14 has been included, for reference purposes, in order to show the ideal attainable V.Es. with different compression ratios, and for different velocities of the incoming mixture in the case of short induction pipes.

It will be seen that the value 80·5 per cent. for the V.E. corresponds with a gas velocity of 100 f.s. in the inlet pipe, and further that the overhead valve type of engine is appreciably the better in V.E.

It should be mentioned that the gas velocities of normal auto-

mobile and aircraft engines varies from about 120 to 180 f.s., but in racing car engines frequently reaches 300 f.s.

(*g*) **Horse-Power Considerations.**—(*a*) The power developed in the cylinder or at the piston is necessarily greater than that at the crankshaft, due to the engine losses.

$$\text{Thus I.H.P.} = \text{B.H.P.} + \text{Engine losses} \quad . \quad . \quad . \quad (1)$$

Further, the ratio $\frac{\text{B.H.P.}}{\text{I.H.P.}}$ is termed the Mechanical Efficiency (M.E.). This ratio is usually denoted by the symbol η.

$$\text{This can be written as M.E.} = \frac{\text{B.H.P.}}{\text{B.H.P.} + \text{Engine losses}} \quad . \quad (2)$$

It follows that in cases where the I.H.P. cannot be measured it is possible by measuring the B.H.P. and also the engine losses to arrive at both the I.H.P. and the M.E.

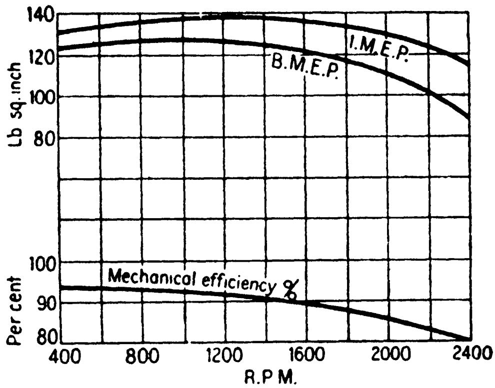


FIG. 15.—Mechanical efficiency and engine speed.

The mechanical efficiency of an engine decreases as the speed is increased, due to the increase in both the mechanical and pumping losses. Both the indicated and brake m.e.ps. show a falling off in value after a certain speed has been attained, but as the brake m.e.p. falls off at a greater rate than the indicated m.e.p., it follows that the mechanical efficiency will diminish with speed increase. These results are illustrated in the case of some tests made by Ricardo, reproduced in Fig. 15.

(*b*) *Calculation of H.P.*—If the indicator diagram is available, the I.H.P. may be computed by measuring the area of the diagram, either with a planimeter, or by the ordinate method, and dividing by the stroke measurement, in order to obtain the mean pressure value.

Alternatively the mean pressure may be measured directly, as will be shown later. Denoting the m.e.p. by p_m lb. square inch, the piston bore and stroke by d and l inches respectively, the r.p.m. by N, and the number of cylinders of the engine by n, the horse-power can be calculated from first principles from the relation

$$\text{I.H.P.} = \frac{p_m \cdot \left(\frac{\pi d^2}{4}\right) \frac{l}{12} \cdot \frac{\text{N}}{2} \cdot n}{33{,}000} = 9{\cdot}91664 \times 10^{-7} \, p_m l \text{N} n \, . \, d^2. \quad (3)$$

If the B.H.P. of an engine is known, then the value of the B.M.E.P. can be calculated from the same formula by substituting the B.M.E.P. for p_m. Thus

$$\text{B.M.E.P.} = \frac{\text{B.H.P.}}{9{\cdot}91664 \times 10^{-7}\, lNnd^2} \text{ lb. per square inch,}$$

or B.H.P. = Constant × B.M.E.P. × N.

The maximum attainable values of the indicated m.e.p. are given in Table II for various fuels; the corresponding values of the brake m.e.p. are obtainable by multiplying these by the M.E., i.e. η.

(*c*) *Practical H.P. Results.*—It is useful for comparison purposes, and as a guide to the standard of performance of actual engines under test to know how to estimate quickly the probable output. To this end the results of a large number of modern engines have been analysed, and an experimental formula (but with a rational basis) has been devised to express the B.H.P. Referring to the expression (3) it will be seen that the quantity $\left(\frac{\pi d^2}{4} \cdot \frac{ln}{12}\right)$ represents the piston-swept volume of the engine. Calling this value V, we have

$$\text{B.H.P.} = \frac{\eta p \cdot \text{V} \cdot \text{N}}{66{,}000} = k \cdot \text{V} \cdot \text{N},$$

where k is a *constant*, so that for a given speed the B.H.P. can be expressed in terms of the cylinder volume. The value of k will depend upon the compression pressure, design of engine, nature of fuel used, etc. The results of tests show that in the case of modern high output four cycle engines, it is usual to obtain about 12 to 16 B.H.P. per litre (1000 c.c.) of piston-swept volume at 1000 r.p.m.; for aircraft engines, from 15 to 20 B.H.P. per litre.

The maximum B.H.P., irrespective of speed, so far recorded is about 180 per litre for a racing car engine. The B.H.P. varies with the speed up to the "peak" of the power curve, so that if, for example, the "peak" occurs at 2500 r.p.m., the B.H.P. at this speed should be $\frac{2500}{1000}$ (7 to 10), or 17·5 to 25 per litre.

The output of a modern two-stroke engine varies from about 7 to 10 B.H.P. per litre at 1000 r.p.m.

Another method of expressing the results is in terms of the piston-swept volume per unit time (litres per minute).

Ricardo has shown that if the results of the large number of aircraft and automobile engine B.H.P. tests available be plotted on the "litres per minute" basis, it will be found that the results lie between 5 B.H.P. per 1000 litres per minute, and 10 B.H.P. per

1000 litres per minute.[1] The average value for a compression ratio of 5, and a brake m.e.p. of 110 lb. per square inch is 8·33 B.H.P. per 1000 litres per minute.

Fig. 16 is a reproduction of a standard B.H.P. reference chart, which may be used for comparison of test results. The various values plotted represent the results of a number of tests made upon automobile and aircraft engines, whilst the radial lines represent the practical ideals, based upon Ricardo's proposed standards. Curve A corresponds to a compression ratio of 6, and a brake m.e.p. of 130 lb. per square inch, and gives 10 B.H.P. per 1000 litres per

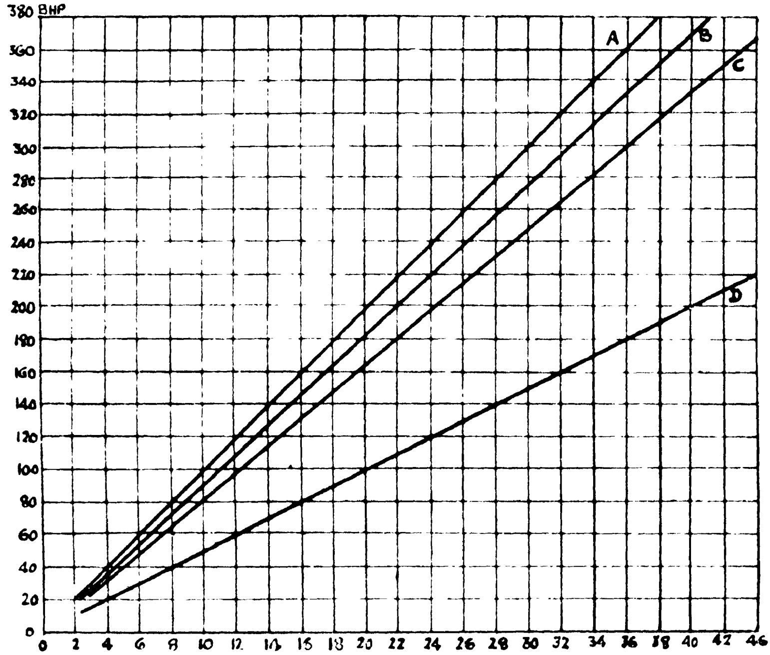


FIG. 16.—Standard B.H.P.-Capacity chart. (Abscissæ represent litres per minute, i.e. cylinder capacity × r.p.m.)

minute. Curve B corresponds to a compression ratio of 5, and brake m.e.p. of 121 lb. per square inch, giving 9·26 B.H.P. per 1000 litres per minute. Curve C corresponds to a compression ratio of 4, and brake m.e.p. of 110, and gives 8·34 B.H.P. per 1000 litres per minute. Curve D gives 5 B.H.P. per 1000 litres per minute.

Compression Ratio and Mean Pressure.—The mean effective pressure of a petrol engine increases with the compression ratio up to the limit of compression fixed by the detonating properties of the fuel employed. As stated previously, fuels of higher octane number are the least prone to detonation, so that if the compression

[1] More recent engines give substantially higher values.

ratio is selected below the limits of detonation for a given fuel the maximum value of the mean effective pressure can be obtained.[1] Fig. 17 illustrates the averages of a large number of tests that have been made upon air and water-cooled engines over a range of compression ratios of 4 : 1 to 8 : 1. These values of the indicated and brake mean effective pressures may be regarded as typical of well-designed petrol engines.

(*d*) *Engine Losses.*—The losses of power in the engine itself may be divided into two classes, namely (1) frictional losses; and (2)

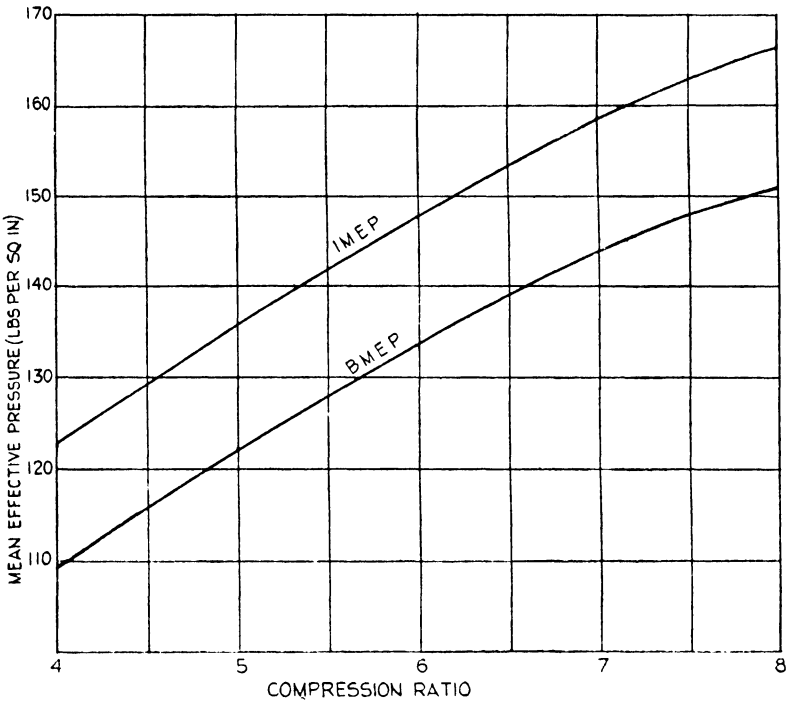


FIG. 17.—Compression ratio and mean pressure losses.

pumping losses. The former include piston and ring friction, friction of the bearings, " air " and crank-case " churning " friction, and power absorbed in driving the " accessories," such as the cooling fan, pumps, magneto, etc.

The pumping losses represent the power used by the engine to draw in the fresh charge and to expel the exhaust gases.

The methods of measuring the losses (1) and (2) will be dealt with in their respective sections later, but a few references will be made here to the magnitudes of these losses in practice.

As a rule the frictional losses constitute about 80 to 85 per cent.

[1] *The Internal Combustion Engine*, D. R. Pye (Clarendon Press).

of the total, but if the volumetric efficiency is low, the pumping losses may be as much as 25 to 30 per cent. of the total. The frictional losses will depend upon the design of the engine, and also upon the lubrication system employed.

Of these losses, that of piston friction is by far the most important; it comprises the friction of the piston surface and that of the piston ring. The following formula expresses empirically the piston friction in lb. per square inch of piston area when the piston length is equal to its diameter:—

$$P_F = 0{\cdot}1(0{\cdot}25P_A + 0{\cdot}66P_1) + C$$

where P_F = piston friction, P_A = mean gaseous pressure, P = mean inertia pressure, and C = a constant which includes the average pressure on the cylinder walls due to compression, and also the effect of the number and strength of the rings. Its value varies from 1·5 to 4·0 lb. per square inch, but may be taken as being about 2·0 for average purposes.

The frictional losses due to the piston depend also upon (1) the nature of the lubricant, being greater for those oils which are most viscous at the working temperature, (2) the temperature of the cylinder walls, and (3) the piston rubbing speed; the losses go up with the speed. Aluminium slipper pistons give appreciably lower values for the piston friction, namely, about 80 per cent. at low speeds down to 60 per cent. at high speeds.

The piston friction expressed as above in a well-designed engine should not exceed about 4 per cent. of the m.e.p. for a piston speed of 600 f.m., and 6 per cent. for 2500 f.m.

The *total* mechanical losses for a well-designed engine vary from about 5 to 7 per cent. of the m.e.p.

The pumping losses in modern engines range from about 2 lb. per square inch of piston area for gas velocities of 100 f.s. up to 9 lb. per square inch for gas velocities of 250 f.s.

The total losses (mechanical plus pumping) in modern petrol engines range from about 7 to 15 per cent. of the I.H.P. With a well-designed engine the mechanical efficiency will often exceed 90 per cent.

Fig. 18 illustrates the mechanical efficiencies of a petrol, gas, and low speed Diesel engine, expressed in terms of piston-speed.

The petrol engine had a bore of 7·25 inches and stroke of 8·5 inches, and gave an m.e.p. of 130 lb. square inch, with a mean gas velocity of 130 f.s. The Diesel engine had a bore and stroke of 16 and 19 inches respectively, normal speed of 250 r.p.m., and mean gas velocity 150 f.s. Its m.e.p. was 89 lb. square inch under these conditions.

Concerning the pumping losses, these are found to vary with the gas velocity through the valves, are obtained from measurements

taken from light spring diagrams of the loop area enclosed between the exhaust and suction lines. The corresponding inlet pressure drop in modern engines varies from about 0·5 lb. square inch for a gas velocity of 100 f.s. up to 2·0 for one of 250 f.s. The mechanical efficiency has been shown to depend upon the piston speed. It also depends upon the following factors :—

(*a*) The diameter of the cylinder, being greater for larger diameters, Callender's formula for the M.E., as modified to suit modern designs is

$$\text{M.E.} = 0{\cdot}98\left(1 - \frac{0{\cdot}3}{\text{D}}\right)$$

where D = diameter in inches.

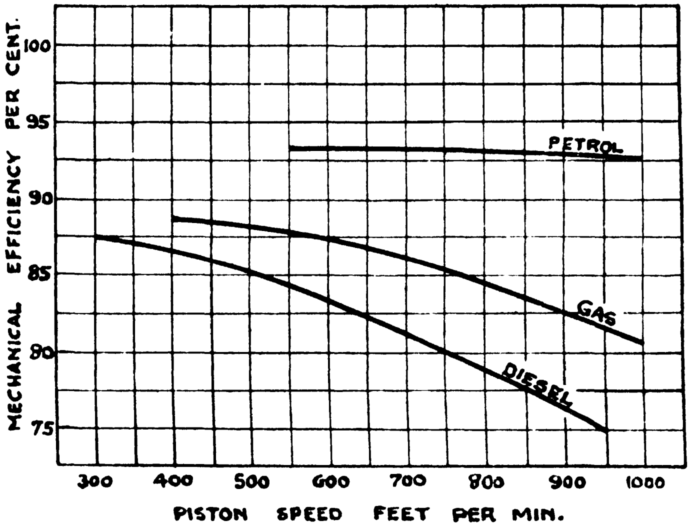


FIG. 18.—Comparative efficiencies of engines.

(*b*) The temperature of the cylinder walls. Its value increases progressively as the cooling water temperature up to about 180° to 200° F.

(*c*) The design of the engine. The highest M.Es. are realized in the case of engines having light pistons, well-lubricated bearings, preferably of the ball or roller type, the minimum of gears and engine-driven accessories.[1] The engine should have relatively light reciprocating and other moving parts, unrestricted valves and ports, and an efficient cam design.

The Variable Compression Engine.—In research work, and also in connection with new designs of high speed engines, the question of compression ratio is important. A number of factors such as the thermal efficiency, the power output, and the tendency to detonate, etc., of different fuels, are dependent upon the compression ratio.

[1] It is usual to include the power required to drive the water-pump, oil-pump, and magneto in the mechanical losses.

It is necessary, therefore, to be able to vary the compression ratio in order to study its influences upon these factors. In the case of fuel research work, in which fuels and fuel mixtures have different limiting maximum compression pressures and ranges, it becomes essential to provide a means for varying this pressure.

With standard engine designs, it is possible to vary the compression ratio, either by fitting pistons of different crown-to-gudgeon-pin lengths, or by the employment of packing liners between the cylinder holding-down flanges and the crankcase. In the latter case it is necessary to provide for the different valve tappet clearances, and sometimes to fit longer holding-down studs.

In the case of a sleeve-valve engine it is not a difficult matter to provide for an up and down adjustment of the cylinder head; there are no valves to consider in this case. With two-cycle engines the whole cylinder head can also be made adjustable, by a screwed barrel or other means.

One of the best examples of a variable compression engine is that employed by Ricardo, in connection with his researches upon various fuels for internal combustion engines.[1] The design of this engine was carefully considered, and a number of very original features embodied. The results obtained from this variable compression engine may be considered as representing about the best possible performance of the four-cycle type on account of the attention given in the design to the reduction of engine losses, maximum combustion chamber efficiency, durability, and reliability.

The engine is shown in section in Fig. 20, in side and front elevations. The method adopted to vary the compression consisted in moving the whole cylinder with its water-jacket and valves in an axial direction by means of a screw thread on the lowest part of the cylinder barrel, engaging with a large " nut," provided with bevel teeth on its lower side. A smaller bevel gear-wheel, actuated by the hand-wheel shown on the right-hand side of the left view, engaged with this larger bevel-wheel, so that the " nut " could be rotated in its plain bearings, thus screwing (without rotation) the cylinder unit up or down. In this manner the compression ratio could be varied from 3·7 to 8·0, whilst the engine was running at full power.

Overhead valves, in the head of the cylinder, were employed, the combustion chamber being practically cylindrical in shape. With this form of combustion chamber the ratio of surface to volume does not vary to any great extent, as the compression ratio is varied. For a compression ratio variation of 3·7 to 8·0, the corresponding surface-volume ratio variation is from 1·6 to 2·75. The bore and stroke of this engine were 4½ inches and 8 inches respectively; there

[1] " The Influence of Various Fuels on the Performance of Internal Combustion Engines," Part VII, *The Automobile Engineer*, August, 1921. See also *Engineering*, September 3rd and 10th, 1920.

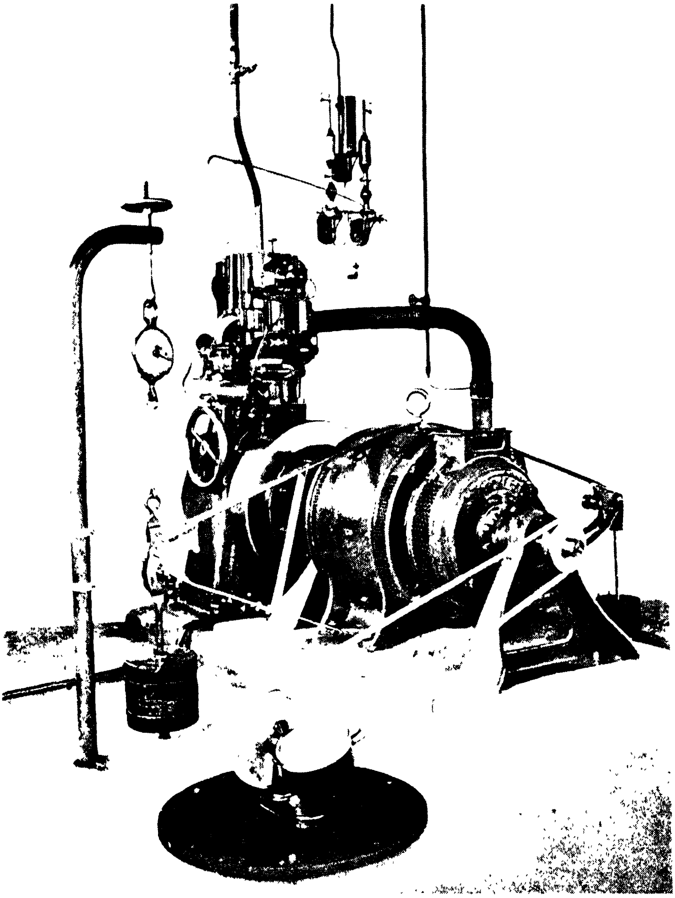


FIG. 10. The Ricardo variable compression engine testing plant.

[To face page 32.

The piston was capable of standing a working pressure of 800 lb. square inch. The connecting rod was of hollow tubular form, of heat-treated nickel-chrome steel. No split big-end bearing was used, on account of the weight, but in place of this a floating bush was employed, running on a case-hardened steel crankpin ; the crankshaft was built-up. The big-end eye was provided with a case-hardened steel bush, and the thin, perforated phosphor-bronze floating bush mentioned previously. Oil under a pressure of 45 lb. square inch was forced continuously through the crankpin by means of a gear-pump fitted in the base chamber ; the oil thrown out from this served to lubricate the piston, gudgeon pin, and other working parts, except the valve gear.

In order to obtain the maximum volumetric efficiency ample valve area was provided, great care being taken to ensure that the best possible valve setting was used ; special attention was also paid to the streamlining of the valve ports and passages, and to the elimination of any abrupt changes in direction or velocity. Finally, the valve seats were masked, or recessed back, the diameter of the recess being only a few thousandths of an inch greater than that of the head of the valve, so that during the first and last portions of its travel the valve acted as a piston valve and gave a sharp cut-off. The valves were made of 3 per cent. low carbon nickel steel, case-hardened all over ; the highly-polished stems ran in close-fitting phosphor-bronze guides.

The valves were operated from an overhead camshaft, each valve having its own independent cam.

The mechanism employed for varying the compression previously mentioned was as follows : The outside surface of the cylinder water-jacket was machined and ground ; it could slide in a massive cast-iron guide. It could be locked by means of a clamp bolt which closed in the guide and so gripped the cylinder uniformly around the whole circumference of the water-jacket. The lower end of the jacket was screwed externally to receive a large phosphor-bronze nut located between the lower face of the guide and the base-chamber. This nut could be rotated by a hand-wheel through the medium of bevel gearing, and the cylinder raised bodily or lowered thereby. The nut and thread were amply strong to withstand the explosion pressures, but to ensure absolute rigidity the clamp bolt in the guide was always tightened up.

A micrometer with electrical contacts, operating a small 4-volt lamp, was fitted to indicate the exact position of the cylinder in its guide ; from this position the compression ratio was known by previous calibration. The valve gear was driven by means of a splined and telescopic vertical shaft. The total available travel was about 2 inches, corresponding to a compression ratio variation of 3·7 to 8·0.

It was possible, with this engine, to run for long periods at piston-speeds of from 2000 to 3000 feet per minute.

Special means were adopted to render the engine independent, as far as possible, of changes in the temperature of the lubricant. Ball-bearings were used whenever possible, in order to reduce frictional variation with different oil temperatures. The water-jacketing around the cylinder barrel was stagnant, so that it quickly attained its normal temperature, which was independent of that of the supply. The piston friction, which is dependent upon the temperature of the oil on the cylinder walls, reached its minimum value in the course of a few minutes and remained at this value thereafter.

Four sparking plugs were fitted at equal distances around the periphery of the combustion chamber; each was connected to a Remy high-tension coil. The low-tension circuit of all the coils was operated by a single Remy contact breaker driven directly from one end of the camshaft; this arrangement ensured absolute synchronism of the sparks at the four plugs, and equal intensities at all settings. It was found that the use of two sparking plugs, on opposite sides of the combustion chamber, gave results equally good with those obtained from four plugs, so that the former arrangement was adopted finally. It was possible in this engine to disconnect either the one inlet or exhaust valve, or both, in order to study the effect of turbulence, valve port area, gas velocity, and other factors. The carburettor, which was attached to the cylinder-head and moved up and down with it, was an ordinary standard Claudel Hobson one, provided with an adjustable needle valve controlling the jet and an electrical heater fitted in the air intake pipe capable of absorbing a maximum of 2·5 kilowatts; this quantity is sufficient to vaporize the whole of the fuel and to raise the inlet air temperature 50° C. if required. The heater was controlled by means of a fine adjustment rheostat, and was in series with the armature circuit of the swinging-field type dynamometer used for measuring and absorbing the power output of the engine.

A revolution counter driven from the camshaft could be thrown in and out of operation by means of a magnetically actuated clutch, which, in turn, was operated from a switch on the fuel-measuring device, so that the actual number of cycles corresponding to the consumption of a given quantity of fuel was recorded automatically.

Fig. 19 illustrates the external general arrangement of the variable compression engine, and shows the swinging field dynamometer with its dashpot (on the right-hand side) and weighing arm (on the left), the fuel-measuring device above, and the belt-driven tachometer immediately in the foreground.

The following results of the calibration tests given in Table V were obtained with a compression ratio of 6, on "Shell Borneo" petrol, over a speed range of 700 to 2300 r.p.m.:—

TABLE V

A.—*Calibration Test Results of Variable Compression Engine*

R.P.M.	B.H.P.	I.H.P.	Friction H.P.	Mechanical Efficiency. Per Cent.	Brake M.E.P. Lbs. sq. in.	Indicated M.E.P. Lbs. sq. in.	Piston Speed. Feet/min.	Gas Velocity through Inlet Valve. Feet/sec.
700	13·7	15·2	1·5	90·0	121·5	134·4	933	51·5
900	17·9	19·9	2·0	90·0	123·0	136·5	1200	66·1
1100	22·0	24·5	2·5	89·8	124·0	138·3	1466	81·0
1300	26·2	29·4	3·2	89·3	125·0	140·5	1733	95·5
1500	30·4	34·5	4·1	88·3	125·5	142·5	2000	111·0
1700	34·6	39·8	5·2	87·0	126·0	145·5	2266	124·8
1900	38·6	45·1	6·5	85·5	126·0	147·7	2533	139·6
2100	42·0	50·3	8·3	83·5	124·5	148·5	2800	154·5
2300	44·3	54·9	10·6	80·7	119·7	148·0	3066	169·0

These values, and also those illustrated graphically in Figs. 21, 22, 23, and 24, have been included in this section to enable

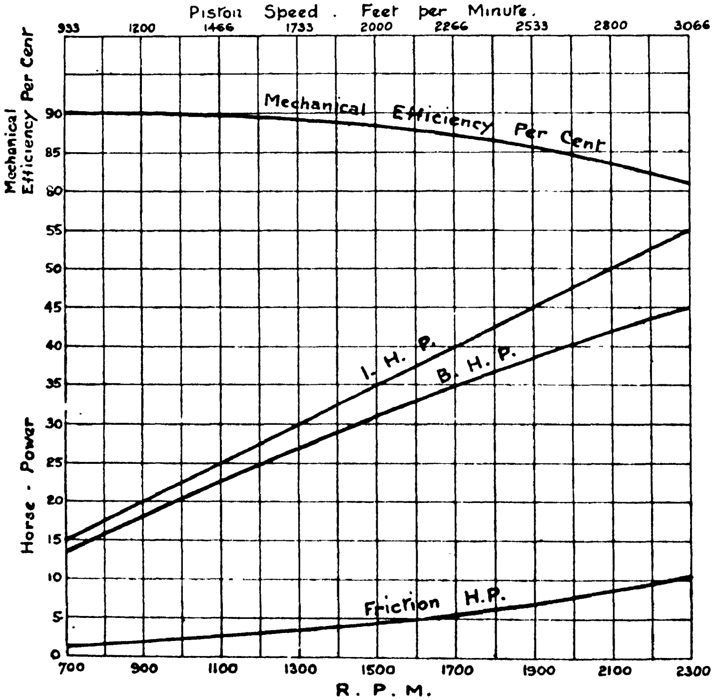


FIG. 21.—Curves showing calibration tests of engine on "Shell Borneo" petrol (compression ratio 6 : 1).

direct comparisons to be made with results obtained from any other engines. Bearing in mind the high standard of efficiency of this engine, the results obtained may be regarded almost as standard

ones, for comparison purposes. Fig. 22 illustrates the efficiency and indicated m.e.p. values obtained with benzole, of calorific value 17,330

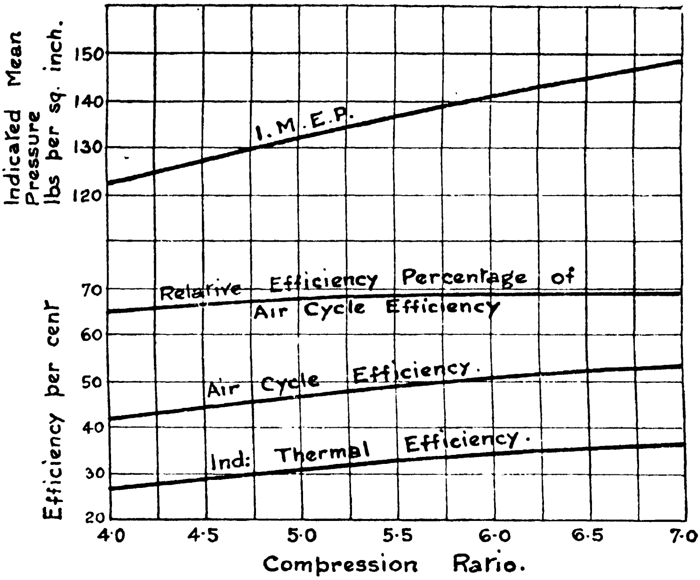


FIG. 22.—Curves showing I.M.E.P. and efficiencies with varying compression ratios.

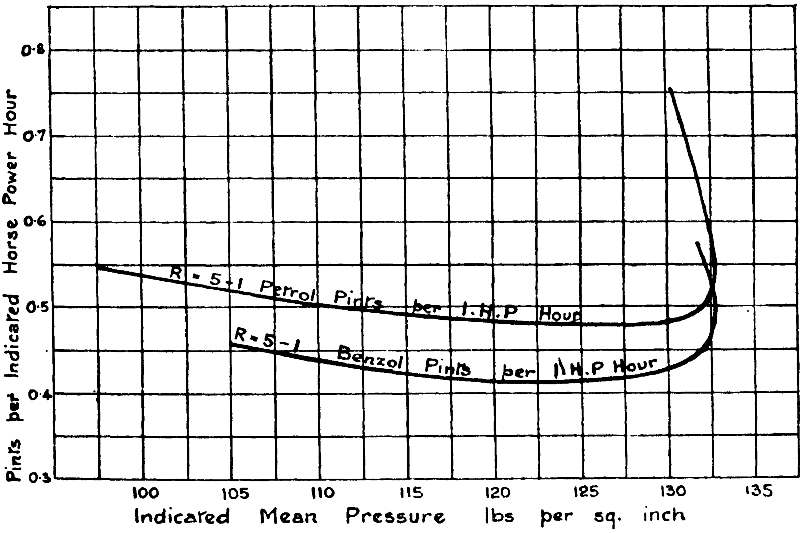


FIG. 23.—Curves showing relation between fuel consumption and indicated mean pressure.

B.T.Us. per pound (19,100 per pint). Up to certain limiting compression ratios, and for the same inlet temperature, the values shown are equally applicable for any fuel.

It will be observed that the efficiency relative to the air cycle rises with increase of compression, and also that there is no corresponding increase in the indicated m.e.p., the rise being considerably less than if proportional to the air cycle efficiency. The relation between the fuel consumption and the indicated m.e.p. is shown graphically in Fig. 23, for both benzole and a light petrol, for a compression ratio of 5. The curves show that the point of maximum economy and maximum torque are very nearly coincident, more particularly on petrol. This is characteristic of all volatile fuels, but does not apply to coal-gas. The general form and character-

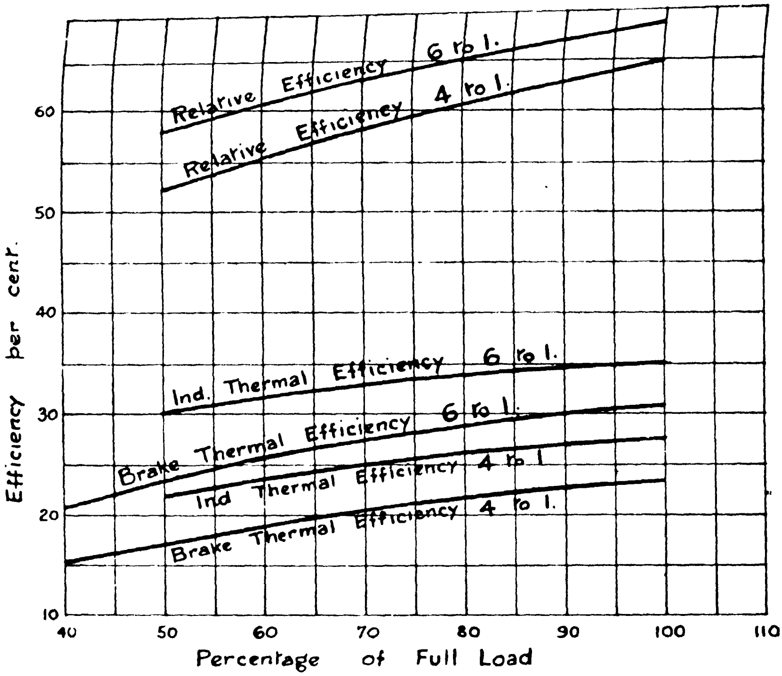


FIG. 24.—Curves showing brake and thermal efficiency in terms of per cent. full load.

istics of these curves are applicable to any fuel and at any compression ratio.

The effect of throttling the engine from full load down to 40 per cent. full load, and for compression ratios of 4 and 6 is shown clearly in Fig. 24. It will be observed that, at very small throttle openings, the results are less uniform ; the available range of mixture narrows rapidly as the engine is throttled down. The curves shown in Fig. 24 have been drawn for the most economical mixture in each case.

The results show that reducing the brake horse-power to 40 per cent. by throttling has the effect of reducing the brake thermal efficiency to 70 per cent. of the full-load value for the compression

ratio of 6, and to 65 per cent. for a ratio of 4. On full load the gain in efficiency from the higher compression ratio is 31 per cent., whilst at 40 per cent. full load the gain is actually 42 per cent.

Standards of Performance.—Ricardo has evolved a standard of attainable efficiencies for four-cycle petrol engines, in which the ideal values are corrected for known effects, and by known amounts. These effects include changes in specific heats, direct cooling, early exhaust, and delayed ignition.

The air-standard efficiency of the ideal engine having the same compression ratio r as the one under consideration is given by

$$E = 1 - \left(\frac{1}{r}\right)^{0 \cdot 408}.$$

Its values for compression ratios of 3, 4, 5, 6, 7, and 8 are 36·1, 44·5, 49·2, 54·8, 57·2, and 58·2 per cent. respectively. The highest values for the corresponding measured thermal efficiencies of petrol engines working on the same four-cycle are very much lower than these, namely, about 60 per cent.

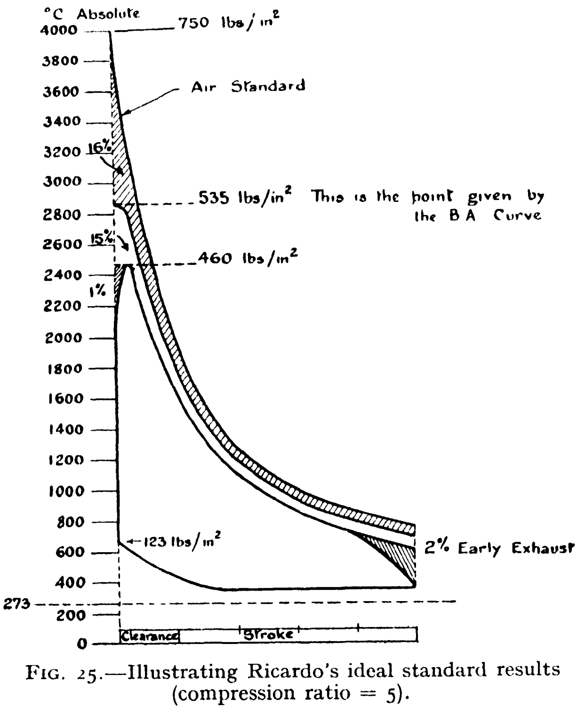


FIG. 25.—Illustrating Ricardo's ideal standard results (compression ratio = 5).

Ricardo has suggested that the ideal air-standard efficiency should be modified as follows, in order to obtain the ideal attainable thermal efficiency :—

Least loss due to change of specific heat	=	16	per cent.
,, ,, ,, ,, direct cooling .	=	15	,, ,,
,, ,, ,, ,, early exhaust .	=	2	,, ,,
,, ,, ,, ,, delayed ignition .	=	1	,, ,,
Total heat loss .		34	,, ,,

Hence the maximum attainable efficiency is taken as 100—34, or 66 per cent. of the ideal value ; this value agrees with the measured values of the thermal efficiencies of different experimenters. Fig. 25 illustrates the modification of the air-standard diagram for each of the four items enumerated above ; this diagram is not to scale.

From the ideal attainable efficiencies the corresponding ideal attainable values for the indicated m.e.ps., and fuel consumptions, and other quantities, can at once be computed for different compression ratio values. Ricardo has also assigned maximum attainable values for both the mechanical and the volumetric efficiencies,

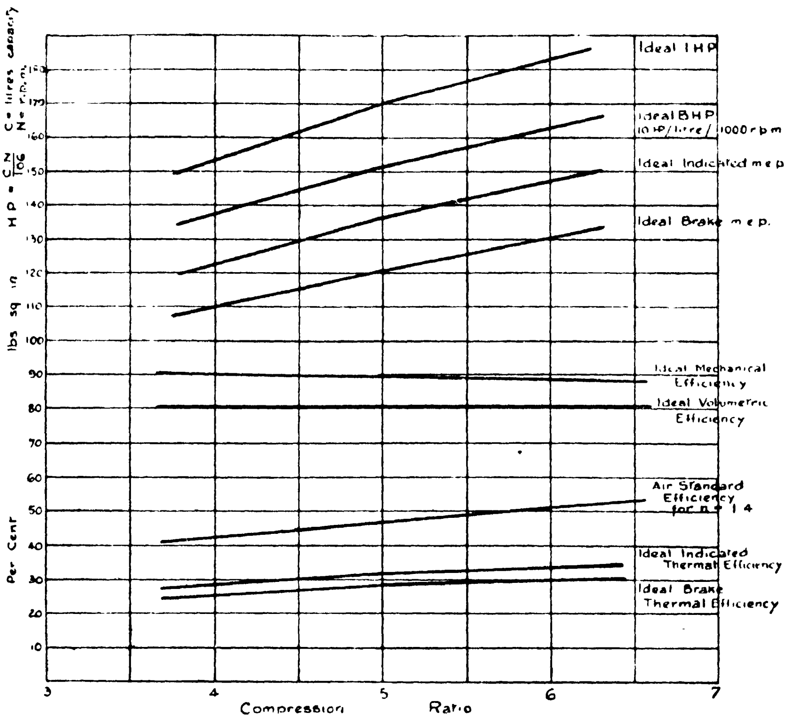


FIG. 26.—Ideal values for different compression ratios.

based upon the results of numerous tests. The volumetric efficiency attainable for a terminal pressure of 14·3 lb. per square inch is given as 80·5 per cent., the loss of charge being analysed as follows :—

Least loss due to heating of gas at inlet valve	=	17·3 per cent.
,, ,, ,, ,, pipe resistance . .	=	2·2 ,, ,,
Total least loss . .		19·5 ,, ,,

This corresponds to a gas velocity of 100 feet per sec. through short induction pipes.

TABLE VI

Ricardo's Proposed Standard of Attainable Ideal Efficiencies for "Four-Cycle" Petrol Engines

Compression ratio r =	4	5	6
Least loss due to change of sp. heat % air standard %	16	16	16
„ „ „ „ direct cooling „ %	15	15	15
„ „ „ „ delayed ignition „ %	1	1	1
„ „ „ „ early exhaust „ %	2	2	2
Ideal thermal efficiency „ %	66	66	66
The air-standard efficiency ($\gamma = 1{\cdot}4$) % absolute %	42·6	47·5	51·1
Ideal indicated efficiency % absolute %	28·2	31·3	33·8
„ mechanical efficiency %	90	89·5	89
„ brake efficiency % absolute %	25·4	28·0	30·1
Ideal petrol consumption ($14{\cdot}5 \times 10^6$ ft.-lb./lb.) Pts./B.H.P./Hr	·592	·535	·50
Ricardo's proposed fresh charge temp. °C. Ab.	330	330	330
Density =	·827	·827	·827
Volumetric efficiency for terminal atm. P = 14·7 lb./in.^{2} %	82·7	82·7	82·7
Ideal volumetric efficiency for terminal P = 14·3 lb./in.^{2} %	80·5	80·5	80·5
Ideal gas energy (petrol C_8H_{18}) ft.-lb./cu. in.	45	45	45
Ideal pump energy (from volumetric effy.)	36·2	36·2	36·2
„ piston energy (from indicated effy.)	10·2	11·35	12·2
„ brake energy (from brake effy.)	9·2	10·15	10·83
Ideal indicated mean pressure = 12 × Piston energy = lb./sq. in.	122·5	136	146·5
Ideal brake mean pressure = 12 × Brake energy = lb./sq. in.	110	121·5	130·5
Hypothetical mean pressure = 20 r ; (Berriman) lb./sq. in.	80	100	120
Ideal indicated H.P. (from piston energy) * = $(CN/10^6)$ ×	153	170	183
Ideal brake H.P. (from brake energy) = $(CN/10^6)$ ×	137·5	152	163
Hypothetical H.P. (Berriman) 25 $CNr/10^6$ = $(CN/10^6)$ ×	100	125	150
Ideal > Hypothetical H.P. %	37·5	21·6	8·7
Hypothetical brake energy (Berriman) ft.-lb./cu. in.	6·62	8·26	9·94
„ brake efficiency (% ideal pump energy) %	18·3	22·8	27·4
„ petrol consumption ($14{\cdot}5 \times 10^6$ ft.-lb./lb.) Pts./B.H.P./Hr.	·81	·65	·54

* C = stroke volume in cu. in. for n cylinders. N = r.p.m.
r = compression ratio.

The mean pressure values corresponding to the ideal attainable volumetric and thermal efficiency values are worked out on the assumption that 1 lb. of petrol has a calorific value of 18,290 B.T.Us., and that 15 lb. of air are required for complete combustion.

The heat energy of this standard mixture works out at 100 B.T.Us. per cubic foot, or 45 ft.-lb. per cubic inch.

The values given in Table VI have been worked out on Ricardo's proposed ideal attainable values, for compression ratios of 4, 5, and 6, respectively. These tables will be found most useful when analysing the performance results of four-cycle engines.

CHAPTER II

TEST PROCEDURE

THE method of arranging and carrying out engine tests depends, as we have already mentioned, entirely upon the nature of the tests to be made. Confining our attention for the present to the three general types, or classes, of test enunciated in the previous chapter, and leaving any discussion of details to subsequent chapters, it is proposed to refer, firstly, to the more elementary tests, and finally to the more advanced tests, namely, those of a research nature.

Bench Tests.—This class includes the simpler kinds of test, such as those of motor-cycle units, and also the more elaborate ones carried out by the manufacturers of high-grade automobile and aircraft engines.

These tests are made with the object of determining the output, fuel and oil consumptions, and general reliability, before the engines are fitted into their frames, chasses, or aircraft.

In the case of newly assembled engines from the production shops, a record will be available of the various clearances and of the adjustments of the valves and tappets, the valve spring strengths, etc., so that the test assistant can proceed right away with the preliminary tests.

On the other hand, an engine which has been sent by a firm of manufacturers to a testing institution may be completely dismantled before any tests are undertaken. In this case the following items will be carefully measured and recorded :—

1. The main dimensions, such as the cylinder diameter and stroke, connecting rod length, sizes of the bearings, diameter of the valves.
2. Piston clearances, and piston-ring clearances and pressures.
3. Weights of the pistons, connecting rods, valves, fly-wheels, etc.
4. Compression ratio, that is, the ratio of the total volume when the piston is in its nearest position to the crankshaft, or inner dead-centre, to the volume of the combustion chamber. This is usually measured volumetrically, by noting the volume of paraffin or other suitable liquid required to fill the combustion spaces. Calling this volume Vc, and d and l the bore and stroke respectively, we have

$$\text{Compression ratio} = \frac{Vc + \frac{\pi d^2}{4}l}{Vc} = \frac{4Vc + \pi d^2 l}{4Vc} = 1 + \frac{\pi d^2 l}{4Vc}.$$

5. Valve lift diagram for inlet and exhaust valves.
6. Valve spring tensions.

Assuming, however, that the engine is received straight from the production shop, it is first mounted upon a specially designed heavy cast-iron or built-up steel test-bed, having bolting down surfaces and holes corresponding to those in the engine bearers. The metal test-bed should be bolted down to a fairly massive concrete base ; reinforced concrete is preferable in this respect. In the Fiat testing department the concrete base has a bed of sand around it to absorb vibrations.

Exhaust Arrangements.—The exhaust pipes should be led away to the outside of the building, so that the gases are quite clear of the test-chamber. The simplest method, and one which is often adopted in the case of production engine tests, is to employ flexible tube couplings between the engine and the silencer box, so that the engine can quickly be coupled up and uncoupled. Care should be taken that both the exhaust pipe and the silencer dimensions are ample for the purpose, so that the back-pressure is not excessive.[1]

In this respect it may be mentioned that the exhaust pipe diameter for a single cylinder engine should not be less than the valve diameter, but preferably from 1·25 to 1·5 times this value.

The silencer can be a cast-iron or sheet-metal vessel of relatively large capacity, namely, of about 0·05 cubic feet for each (maximum) B.H.P. of engine. For ordinary silencers, provided with the usual baffle arrangements, a silencer capacity of 0·008 cubic feet per B.H.P. has been shown, experimentally, to give satisfactory results. Another and more convenient method of expressing the unbaffled silencer is as follows :—

Capacity of unbaffled silencer in cu. ft.
= 0·8 litre of cylinder capacity.

Thus a four-cylinder engine of 90 mm. bore and 120 mm. stroke, corresponding to a total capacity of 3·053 litres, would require a silencer, of the free-vessel type, of 2·44 cubic feet capacity. Many manufacturers run their exhaust pipes from the engine into a common pipe of large capacity. In the case of the Fiat testing plant, the exhaust pipes, which are quite short, lead into an underground concrete conduit of large diameter, which passes to the rear of the test blocks. This conduit discharges into the open air, and is provided at its extremity with a powerful centrifugal aspirator. In this way there is for each engine on the test beds not more than a foot of visible exhaust piping ; there is a notable absence of noise, fire risks are reduced to a minimum, and the test house has a very clean appearance. The exhaust arrangements adopted in typical cases will be observed from several of the photographs reproduced in this volume. The writer has employed an underground concrete silencer of large capacity, filled with coke,

[1] A spring-loaded flap, to act as a pressure-release when back-fires occur, is an advantage, and will prevent silencer damage.

the exhaust gases entering at one end, passing through the coke pieces (which averaged about 2½ in. in mean length), and out through fairly large openings at the other end. If the path of the gases is thus broken up, and the gases are finally ejected into the open air in a more or less continuous stream or streams, practically no noise will occur.

One of the principal objections to test houses in the vicinity of dwellings is that of the exhaust noise; by adopting one of the silencing arrangements outlined above, this objection may be overcome without introducing any appreciable back-pressure or temperature effects.[1] The stationary gas engine type of silencer may be considered a suitable example to follow.

Some additional information on the subject of silencing aircraft engine test houses is given on pages 375, to 378.

Preliminary Running-In.—A new engine should be given a preliminary running-in period, the length of which will depend upon the manufacturing methods employed. If the cylinders, pistons, and rings are finished by the grinding process, a longer running-in period will be required than if they are lap-finished.

No definite rule can, therefore, be laid down, but so far as car engine practice is concerned, engines of the mass-production type, which frequently are "tight" on assembly, should be run-in at about one-quarter power for about ten to twelve hours, during which careful attention is given to the lubrication, etc. The running-in period is essential in all cases, since it is not possible, commercially, to obtain the same degree of polish and texture of the rubbing surfaces which results after an engine has been in service for the above minimum period. Moreover, the piston-rings are seldom properly bedded in under about twelve hours' running.

The minimum running-in period advisable is about four to six hours, at from one-quarter to half load, during which period a careful note should be made of any abnormal variations in the power, speed, or heating.

One well-known high-grade car-manufacturing firm has laid down the rule that all new engines shall be given a running-in period of one hour under light load, and four hours normal load, on town-gas fuel, in the case of their smaller engines (10 h.p. R.A.C. rating), and two hours' light, followed by seven hours' normal load running for their larger engines (20 to 30 h.p.).

The use of coal-gas for running-in engines is a commendable one. It has been the practice at several works to run their engines in by this means. For nearly all testing, except that of actual power and fuel consumption, the use of coal-gas is beneficial. Not only is there a saving in the actual cost of running, but the existing stocks

[1] *Vide* Figs. 29 and 344 for typical test exhaust arrangements.

of petrol and other fuels are not deplenished. Moreover, the use of coal-gas is particularly adapted to lighter load tests.

The ordinary type of carburettor, with a simple adapter, or nozzle, can generally be used, but it is a straightforward matter to construct a gas carburettor. The proportions of coal-gas to air for satisfactory running range from about 4 parts of air to 1 of gas, to 8 parts of air to 1 of gas. An extra air valve or port, with regulation means, should be provided.

For absorbing the power during these running-in tests there is a choice of several devices, namely, the solid friction type of Prony brake, the fan or air-resistance type, the electric and the hydraulic brakes.

If it is proposed to subject the engine to subsequent tests of an accurate and extended nature, it is usual to employ one of the two latter types of brake, but in the case of only approximate tests, either of the two former methods may be employed. Probably the simplest and most satisfactory method is the fan-brake one, in which a sufficiently large pair of plates are fitted to the fan-brake torque arms to ensure the revolutions attained, or power absorbed, being well below the maximum values. In the case of aircraft engines a more powerful type of propeller than the normal is sometimes employed for certain tests.

Subsequent Tests.—*Production Engines.*—Consider, firstly, the small single-cylinder type of engine, such as the small stationary lighting or power type, the motor-cycle and motor-boat types; the bench tests made include :—

1. Maximum B.H.P. maintained, say, for a period of an hour.
2. Endurance test at normal power, of from twelve to fifty hours as a rule.
3. Fuel consumption tests at normal and maximum power.
4. Oil consumption tests.

Items 1, 3, and 4 can be measured simultaneously by two test assistants, whilst item 2 requires only the attention of one assistant. It is necessary before proceeding with the above tests to carefully check the valve tappet clearances, and to ascertain by trial the carburettor jet, choke and air settings, and ignition timing, for the best performances, on the lines indicated in the preceding chapter, namely, by varying one factor at a time. The engine should be given a preliminary run, until the whole of the metal parts and the lubricating oil have attained their proper working temperatures.

It is necessary to provide the following apparatus for tests of this nature, viz. a power brake or dynamometer, tachometer, revolution counter, and stop-watch (for checking the tachometer), fuel-measuring device or flowmeter, oil-measuring device (a graduated vessel is all that is required), and an ordinary thermometer for air temperature. In the case of water-cooled engines, the temperature

of the inlet and outlet water, and the quantity of water circulated may be ascertained; from these measurements useful data on the cooling can be obtained.

The following table (p. 48) is a reproduction of a typical test chart of the kind used for the more simple bench tests. Its arrangement ensures the noting down of all of the essential measurements, whilst the final results can be calculated and filled in in the columns provided.

It will be observed that a column is included for the oil consumption. The crank-case oil sump, or the separate oil reservoir, as the case may be, is filled up with oil to a fixed definite level, preferably up to a pointed needle or rod, which just touches the surface, as viewed with a small pea-lamp. At the conclusion of a certain period of running, the level is restored by pouring (warmed) oil into the reservoir until the surface just touches the needle; the quantity of oil thus poured in represents the oil consumption in the period intervening. An ordinary oil gauge glass can of course be used.

No column has been given for quantity of water, the temperature only being taken. Further, the tachometer reading is used only for rough speed indication, the true speed being obtained from the difference between the total revolution counter readings, divided by the time in minutes.

At the conclusion of the performance and duration tests, the engine should first have the tappet clearances, etc., checked, and a note made of the carburettor settings and ignition timing, etc. It should then be dismantled, thoroughly examined, and clearances checked. After cleaning, it should be re-assembled, fresh oil put into the sump, and a further test run given of an hour or two, before being given its final road test on the car or cycle, if an automobile engine.

Tests of New Type Engines.—In the case of experimental or new designs of engines, as, for example, the engine for a subsequent car model, or for special competition purposes, a rather more stringent test is necessary if the best results are to be obtained. It is not only necessary to measure the B.H.P., fuel and oil consumptions, but also to investigate the sources of power loss, and to analyse the performance results with a view to introducing design improvements. It is now fairly well known, as a result of numerous scientific tests which have been carried out in the past, what the output, endurance limits, fuel and oil consumptions of an engine of given dimensions and compression ratio should be. Indeed, we now have accurate standards of comparison, from which the performance of any new engine of the standard type can be judged.[1]

There is, however, the constant search for new types of engine,

[1] *Vide* p. 39.

LOG SHEET A

Example of Test Sheet Heading (Bench Test)

Manufacturer's Name. .... No. of test........

Engine model....Bore....Stroke....Log Sheet No.........
No. of cylinders........ Compression ....
Cylinder capacity.............. Date................

No. of Observation	Time	Tachometer Reading	Rev. Counter Reading	Revolutions per Minute	Dynamometer Reading	Brake Horse-Power	Temperature of Jacket Inlet Water	Temperature of Jacket Outlet Water	Temperature of Room	Fuel Readings. Time per given Vol.		Fuel per Hour	Fuel per B.H.P. Hour	Vol. of Oil to Maintain Oil Sump Level	Oil per B.H.P. Hour
										Start	Stop				
	H.M.S.	R.P.M.					° Fah.	° Fah.	° Fah.	M. S.	M. S.	Pints	Pints *	Pints	Pints

* To convert to lbs. multiply by 1·25 × S.G. of fuel.

and for improvements in the standard type, but here again the performance of the existing standard engine type can be utilized for comparison purposes.

Upon the receipt of a new engine in the test laboratory of the works, a record should be made of its dimensions, clearances, moving parts weights, compression ratio, and any other special information. This information, in the case of a manufacturer testing his own products, can be ascertained during the design and constructional stages.

If, however, the engine is made by contract, or sent to an independent test laboratory, it is the usual procedure to dismantle the engine, measure the clearances, and take its dimensions and weights, to examine the workmanship and fitting, making good any errors found, to thoroughly clean and assemble it, and adjust the valve tappets to the best clearances dictated by experience for this type of engine.

Fresh oil is put into the sump before running the engine. The valve lift and opening diagrams are obtained by means of a suitable indicator. The flywheel rim is usually marked off either with the dead centre and valve opening and closing positions, or, better still, in degrees, and with the dead centres indicated ; a fixed pointer on a convenient part of the crank-case is used as a reference mark.

Carburettor Alterations.[1]—For the purposes of the test it is usual to fit a modified type of the standard carburettor, for convenience of mixture adjustment, and for obtaining a high standard of comparison.

The carburettor jet, if not already adjustable, is fitted with a needle valve, operated by a milled-headed screw, graduated in tenths, and a vertical scale ; this enables every setting to be recorded. Automatic mixture controls must be eliminated for these tests, since it is required to maintain the mixture constant at different pre-arranged values.

The carburettor choke tube is fitted which gives approximately 2½ times the gas velocity through the inlet valve. The throttle should be examined to make sure that there is no wire-drawing of the mixture. For this reason the cylindrical barrel type of throttle is preferable to the more widely used butterfly-valve type.

The air supply to the carburettor should be arranged so that it can *all* be drawn from one source, namely, an air-flowmeter, or calibrated gasometer.

The heating of the carburettor is an important item in engine tests, since any heat supplied to the mixing chamber, as in the case of hot-water or exhaust-jacketed carburettors, has a marked influence upon the volumetric efficiency and, therefore, on the m.e.p.

[1] *Vide* also "Motor Manuals," Vol. II, *Carburettors and Fuel Systems*, A. W. Judge (Chapman & Hall, Ltd.).

Ricardo recommends the fitting of an electric resistance heater to the air intake, the current and voltage being measured ; in this way an accurate measure of the watts provided per given quantity of air flowing into the carburettor is obtained.

Fig. 27 illustrates the effect of the carburettor heating upon the m.e.p. and volumetric efficiency ; the results show a progressive diminution in these quantities with increase in carburettor heat.

All possible sources of air leakage, such as those past the throttle ends, the valve stems, and inlet pipe joints should be examined, and any leakage stopped with insulating tape, Chatterton's compound or asbestos string.

Water Connections.—In some cases, particularly for special commercial engine tests, a large capacity radiator, provided with an engine-driven cooling fan, is supplied. The cooling water is circulated around by the engine's own pump, and its inlet and

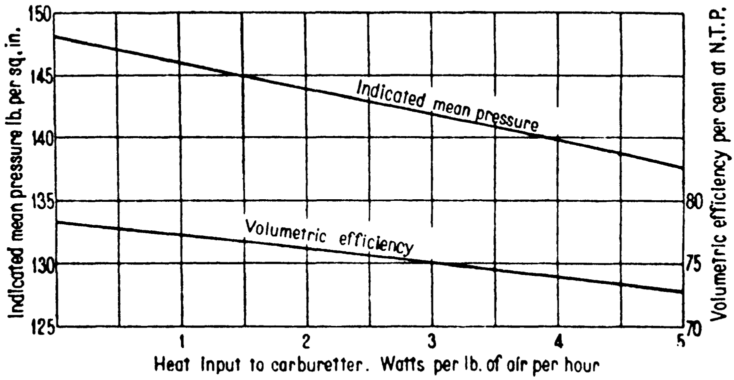


FIG. 27.—Effect of carburettor heating upon the i.m.e.p. and volumetric efficiency.

outlet temperatures are measured. This method possesses the drawback that the actual quantity of circulating water cannot be measured very accurately ; in many cases this is, of course, not necessary.

For other purposes it is necessary, however, to know the amount of heat carried away by the cooling water, and also to be able to control the temperature of the water. It is not proposed to discuss the methods of measuring the quantity and rate of water circulation here, but rather to point out the necessity of providing adequate and controllable water circulation means.

Thermometers should be provided at the inlet and outlet of the cylinder jacket ; these may be of the mercury-in-glass or of the distant reading vapour-pressure type.

Apparatus for Special Engine Tests.—The following apparatus is necessary for the testing of special engines, in order to

obtain all of the strictly essential data required for a true conception of its practical and comparative merits :—

(1) A rigid test-bed, with convenient brackets and fittings for certain of the accessories and test apparatus. (2) An accurate form of dynamometer for absorbing and measuring the power. In this respect one with a variable load device is preferable. If of the electrical swinging field type, the connections can be so arranged that the dynamo can be converted into a motor for starting up the engine, and for motoring the engine around in order to ascertain its losses. (3) An air-measuring device, for accurately determining the quantity of air consumed. One or two alternative methods of air-measurement are described later. (4) An accurate fuel measuring device, such as the graduated vessel, or flowmeter type; it

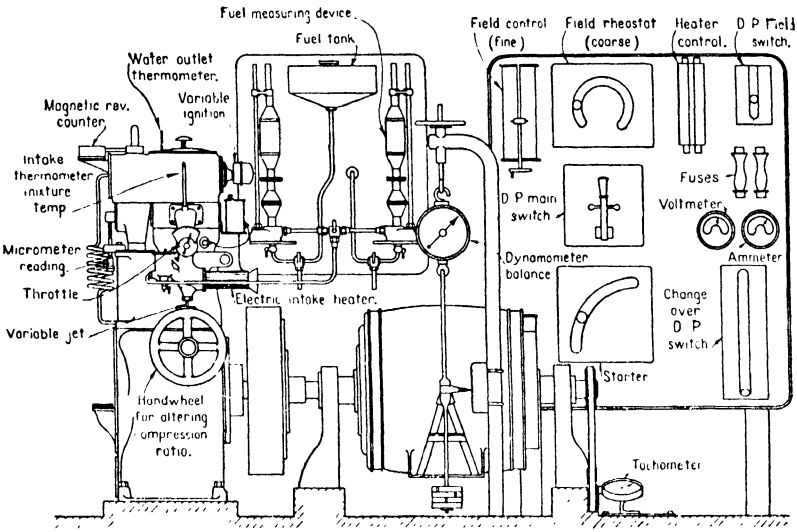


FIG. 28.—Showing diagrammatically the arrangement of the Ricardo testing plant

should be independent of the viscosity of the fuel. (5) A tachometer and revolution counter; one to give the approximate and the other the absolute speed. (6) Thermometers of the mercury-in-glass, vapour pressure, or thermo-couple type; the latter possess the advantage that they can be made distant reading and recording. (7) Jacket water-cooling system, whereby the cooling water can be maintained at any desired temperature. The auxiliary reservoir, air-cooled radiator, or injector methods described later are available for this purpose. (8) Means for supplying a known amount of heat to the carburettor. (9) Manometers or low-pressure gauges for the inlet and exhaust mean pressures, fitted with suitable damping means.

Although the above apparatus will be sufficient for most of the data and information required in the special type of test in question,

namely, that of a new design of engine, yet it is an advantage to be able to ascertain exactly what is going on in the cylinder itself. The compression and explosion pressures can be measured with a special gauge, such as the 'Farnboro' maximum pressure electrical gauge, or the Okill pressure indicator, both of which are described later.

The instantaneous pressures can only be measured by means of special high speed engine indicators now available. With such indicators the I.H.P. can be measured. Since the B.H.P. also can be determined from the brake-dynamometer readings, the total engine losses are determinable directly. Moreover, the pumping losses can be measured from the indicator diagrams (light spring ones) and considerable light thus thrown upon the performance.

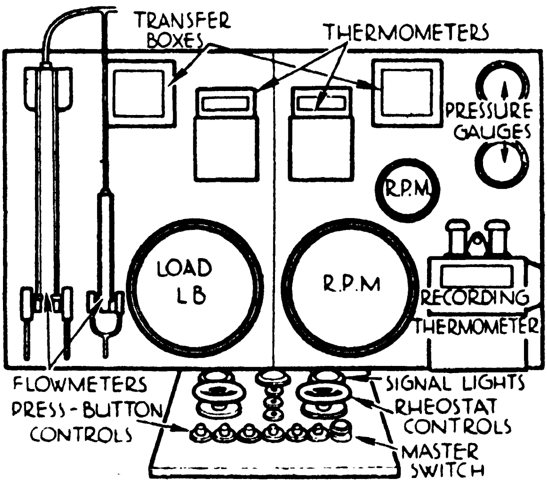


FIG. 30.—A typical engine-power testing control panel.

Fig. 31 illustrates the motor research laboratory of the late Prof. W. Watson, F.R.S., showing an aircraft engine on the test-bed. To the right will be seen the air-boxes and pipes for the air supply measurements. Above is the fuel-measuring apparatus, the glass vessel to the left of the petrol can being used for consumption tests. An optical indicator is shown on the table to the left, with its chain drive for the phase gear. The cooling fan, water circulating pump, magneto, revolution counter, and oil pressure gauge can also be observed.

Fig. 30 shows the arrangement of the instrument and control panel of an engine testing plant built by the Highfield Electrical Co., Ltd., Coventry, for the Automotive Engineering Co., Ltd.

The engine-power testing plant consists of a dual-dynamo torque-reaction dynamometer. The press-button controls and fine-adjustment rheostat wheels are shown in the lower part of the

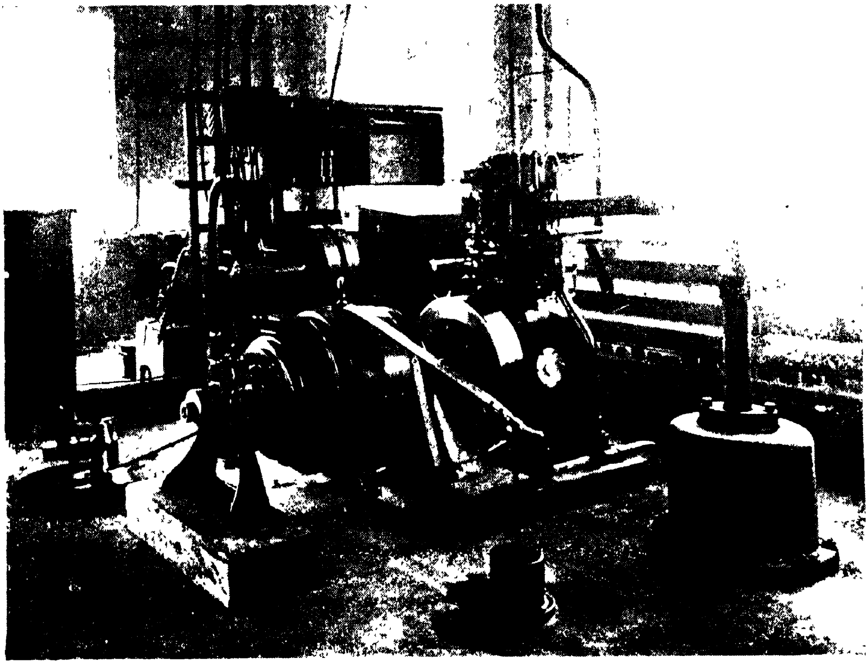


FIG. 20 Ricardo test plant with single cylinder engine unit.

[*To face page 52.*

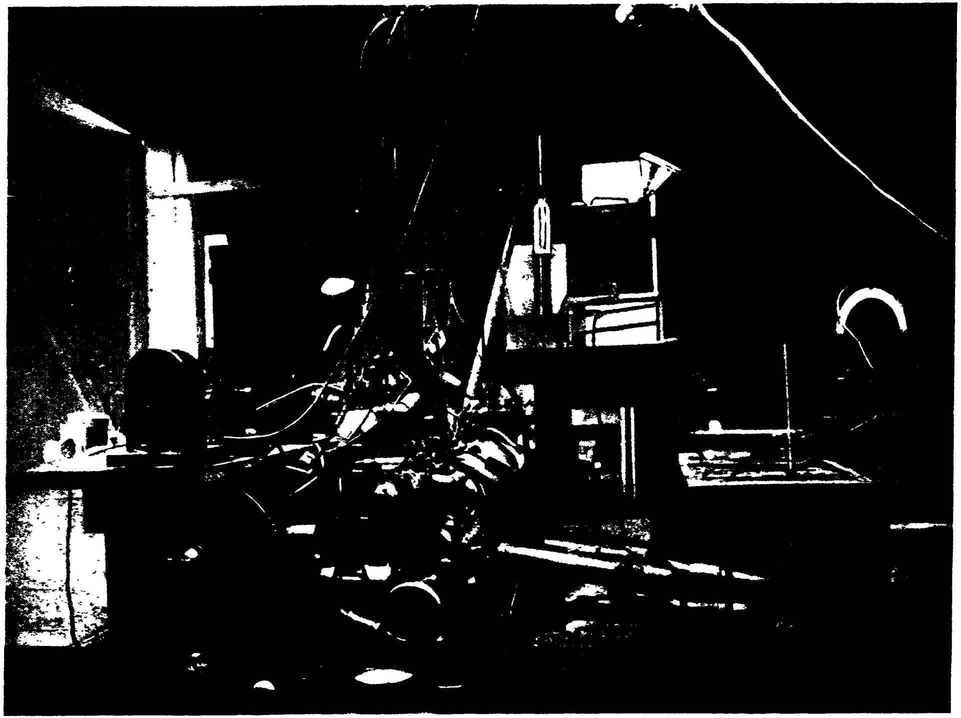


FIG. 31.—The petrol engine research equipment (Royal College of Science).

[*See page* 53.

illustration. A main switch controls the whole circuit, and when this is making contact a red light shows on the control panel. A green light informs the operator when the load is actually placed on the engine. A white light warns the operator when the "motoring" test is in progress, i.e. when the electrical connections have been altered so as to convert the dynamos into motors in order to turn the engine.

Two electrical distant-reading thermometers are provided on the panel, each of which can be connected by means of a switch gear to any one of six points on the engine. One is operated from a thermo-couple for high temperatures up to 700° C., and the other is connected to a platinum resistance and records up to 200° C. In addition, there are two continuously recording devices to which these thermometers can be coupled. When this is done a permanent graphical record is made in ink on paper secured to a drum which is rotated by a clock mechanism.

The panel equipment is completed by a pair of pressure gauges and two flowmeters which record the rate at which petrol is being used by the engine. Consequently, a complete set of test figures for consumption, torque and power at various speeds right through the range can be obtained with great rapidity.

Test Procedure.—Assuming that the new engine has been carefully measured up and re-assembled, the valve clearances having been adjusted, etc., the engine is started up, either by hand or with the electric motor arrangement previously referred to. The engine is run light for about an hour, or until every part, and the lubricating oil, have been thoroughly warmed up to their working temperatures. During this period the working of the various test accessories can be checked, and any minor defects rectified.

It is advisable at this stage to ascertain whether the engine losses are about normal, in comparison with those of standard engines of the type under test. If an electric swinging field type dynamometer is used, the ignition and water circulation device can be switched off, the engine motored around at its previous speed, and the power required to drive it measured approximately. Otherwise the longer process of measuring the B.H.P., by means of the dynamometer, and simultaneously taking indicator diagrams from the different cylinders must be employed.

If the engine losses are about normal, the tests can be proceeded with, otherwise an examination, or a longer running-in period are indicated.

Everything being in order, the engine is run under full load, and the carburettor jet-needle and timing-lever adjusted, so as to give the maximum output. The temperature of the circulating water should be varied and a note made of the temperatures of the inlet and outlet water for the best outputs. A series of horse-power,

fuel, and oil consumption tests can now be made with open throttl
and at various speeds so as to obtain a number of points on th
power and fuel consumption curves. During each test the spee
must be maintained constant, and the best combination of fu
*supply, ignition advance, and circulating water maintained fo
this speed. The speed range of the tests should exceed the spee
range* for maximum power. The mechanical losses can now b
determined at a number of speeds over the previous range. Fo
this purpose the engine is first run under its own power at each speed, and, when conditions are steady, both the circulating water and the ignition should be stopped, and the engine motored around at the given speed. The power necessary to motor it can be measured either from the known electrical efficiencies of the motor, or more conveniently from torque arm measurements, if the dynamometer is of the swinging field type.

One other test is necessary, for what one may term the "minimum requirements" tests, namely, the measurement of the volumetric efficiencies at different speeds. If the quantity of air flowing through the carburettor, per minute say, be measured during the above horse-power-speed tests, the volumetric efficiencies at different speeds is determinable.

From the above tests the following information is obtainable :—

1. The B.H.P. at various speeds up to maximum power output.
2. The I.H.P. as deduced from the B.H.P. and engine losses.
3. The thermal efficiencies at the maximum B.H.P.-speed values. These efficiencies are not the maximum, however, but correspond to the richer mixture values giving the greatest B.H.Ps. at the different speeds.
4. The mechanical efficiencies at different speeds.
5. The volumetric efficiencies ,, ,,
6. The mean torque values ,, ,,
7. The fuel consumptions per B.H.P. hour at different speeds.
8. The best cooling water temperatures for maximum B.H.P.
9. The best ignition settings for maximum B.H.P.
10. The effect of varying the carburettor heat input, as measured from the electrical energy supplied.
11. The oil consumption per B.H.P. hour at different speeds.

It should here be mentioned that the determination of item (3) involves, in addition to the fuel consumption, a knowledge of the calorific value of the fuel.

The items enumerated may be regarded as minimum requirements in the investigation of a new type of engine, and would correspond to the tests carried out by an automobile or aircraft engine manufacturer in his own test laboratory.

Methods of Recording Test Data.—The method to be adopted for recording test data should be carefully considered beforehand,

as otherwise a great deal of inconvenience may result during the tests. Test forms are undoubtedly the most practical and useful means for noting down and recording the results.

Probably the best method is to have all of the required data in a convenient form for making the final calculations, and for filing purposes.

The writer has found the printed test-sheet pad, provided with a good, stout cardboard back, most useful. The various columns are ruled off, and the headings carefully selected and arranged in the order in which the results are obtained. Full particulars of the type, number, dimensions, and other items should be filled in at the head of the sheet, together with the test and filing number. In this manner test sheets can be arranged to contain all of the original readings and data for any future reference or check purposes.

In some cases it is found advisable to detach test sheets from their block, and to fasten them, with spring clips, to a plywood board provided with a pencil (on a chain) and stop-watch holder.

A convenient form of test sheet, similar to that employed at the Ricardo works, is shown overleaf.

One of the simplest forms of test sheet for the present type of tests is probably that shown in the S.A.E. LOG SHEET B,[1] since it enables the B.H.P. and fuel per B.H.P. to be filled in directly.

For more extensive tests it is better to limit the test sheet to test readings only, and to provide a second sheet for the whole of the engine specification data, as in LOG SHEET C, and a third sheet for the worked-out results. It is, of course, possible to so arrange the test sheet that both the readings and the final (calculated) results are all contained on the same sheet, similar to the S.A.E. LOG SHEET C given in LOG SHEET D, although this gives a rather crowded form.

The final calculations are preferably carried out with logarithmic tables of the four, five, or seven figure types, according to the accuracy required, or alternatively to the same limits as the probable degree of accuracy of the readings.

In all cases the results computed by one observer or assistant should be checked independently by another person; in many cases a slide-rule check is quite useful.

The causes of any exceptional or incorrect values obtained should be enquired into, and, if necessary, the tests repeated. The method of presenting the final results should invariably be the graphical one, where series of observations over a range are taken. For this purpose it is recommended that special squared paper pads be obtained, with the various ordinates and abscissæ, and the quantities to be plotted, printed thereon. A convenient form is that

[1] Society of Automotive Engineers, 29 West 39th Street, New York City, N.Y., U.S.A.

Engine Test Sheet

Sheet No. ....

Test No. ....

Test on .... Date ....

State or any special condition of Engine .... Baro. ....

.... Fuel ....

.... Sp. Gr. .... 15° C.

.... Oil ....

Carburettor .... Settings .... Plugs ....

Time	R.P.M.	Torque ... Ins: Rad			B.M.E.P.	B.H.P.	Fuel Consumption Unit Vol. Pint		Ignition Setting	Induct Temp. ° C.	Water Temp. ° C.	Oil Press	Observers
		W lbs.	W lbs.	Nett Weight			Seconds Unit Vol.	Pint per B.H.P.-hr.	Full ADV.				

LOG SHEET B

(A) *Brake Horse-Power, Fuel, Oil, and Air Consumptions*

Name of Manufacturer.
....
Name of Engine.............. Model No................
No. Cylrs......... Bore......Stroke...................
Cubical Capacity.............. Compression Ratio........
Log Sheet No...........

Fuel.............. Specific Gravity........at........°F.
Calorific Value................B.T.U./lb.
Dynamometer.................... Arm............ft.
Barometer................ Humidity................
Observers...... Room temp. before......°F. ; after......°F
Date............

Run No.	Time	Counter Reading	Tachometer Reading	Total Time	Total Revs.	Mean R.P.M.	Brake load at Arm R.	Torque in lb.-ft.	B.H.P.	Friction H.P. at (N) *	I.H.P.	% Mechanical Efficiency	Inlet Water Temp.	Outlet Water Temp.	Carburettor Air Temp.	Air Flowmeter Reading	Fuel Readings. Time per given vol.		Fuel per hour	Fuel per B.H.P. hr.	Oil supplied to Sump Level	Oil per B.H.P. hr.
																	Start	Stop				
	M.S.		R.P.M.	M.S.		(N)	lb.						° Fah.	° Fah.	° Fah.	cu. ft.	M.S.	M.S.	lb.	lb.	lb.	lb.

(B) *Frictional Horse-Power* †

Run No.	Time	Counter Reading	Tacho-meter R.P.M.	Total Time	Total Revs.	Aver. R.P.M.	Mean Temp. Jacket Water	Break Load Arm Reading	Torque	Friction H.P.	Remarks

* From (B). † This section may be arranged conveniently at the foot of the test sheet.

S.A.E. GASOLINE ENGINE TESTING FORM **LOG SHEET—C No.______**

Name______	Fuel______B. T. U. per Lb.______Sp. Grav.______at______deg. fahr.
Model______No. Cyls.______	Dynamometer______Arm (R)______ft.
Bore______In. Stroke______In. Displ. (D)______Cu. In.	Humidity______per cent.
Laboratory______Date______	Oil______Grade______Cold Test______deg. fahr.
Observers______	Saybolt Univ. Vis. at 130 deg. fahr.______At 210 deg. fahr.______

	RUN NUMBER	SYMBOL	FORMULA	1	2	3	4	5	6	7	8	9	10
BRAKE HORSEPOWER AND FUEL CONSUMPTION	TIME STARTED		•										
	DURATION OF RUN—MIN.	t	•										
	COUNTER START	C_o	•										
	COUNTER FINISH	C_t	•										
	TOTAL REV.	r	$C_t - C_o$										
	AVERAGE R.P.M.	N	$\frac{r}{t}$										
	BAROMETER IN. MERCURY		•										
	ROOM TEMP.		•										
	CORRECTION FACTOR	C.F.	$\frac{P_s}{P_o} \times \sqrt{\frac{T_o}{T_s}}$										
	BRAKE LOAD AT ARM R	P	•										
	BRAKE LOAD CORRECTED	P_c	CF × P										
	TORQUE LB FT.	T	PR										
	BRAKE M.E.P.	$b p$	$\frac{180.8\,T}{D}$										
	BRAKE HP	B. HP.	$\frac{PRN}{5252\ 1}$										
	FRICTION HP. AT N	F. HP.	F.HP. Curve										
	INDICATED HP.	I.HP.	B.HP + F.HP.										
	MECHANICAL EFFICIENCY	M.E.	$\frac{B.HP.}{I.HP.}$										
	TEMP. JACKET WATER—IN		•										
	TEMP. JACKET WATER—OUT		•										
	TEMP. OIL—IN		•										
	TEMP OIL—OUT		•										
	OIL PRESS., LB		•										
	TEMP AIR TO CARBURETER		•										
	WT FUEL START	W_o	•										
	WT. FUEL FINISH	W_t	•										
	LB. FUEL USED	W	$W_o - W_t$										
	LB. FUEL PER B.HP. HR.	F	$\frac{60\,W}{t \times B.HP.}$										
	THERMAL EFF. RE B.HP.	T.E.	$\frac{2545}{F \times B.T.U.}$										
		•											
FRICTION HP.	TIME STARTED		•										
	DURATION OF RUN—MIN.	t	•										
	COUNTER START	C_o	•										
	COUNTER FINISH	C_t	•										
	AVERAGE R.P.M.	n	$\frac{C_t - C_o}{t}$										
	BRAKE LOAD AT ARM R	p	•										
	FRICTION HP	F HP.	$\frac{pRn}{5252\ 1}$										
	MEAN TEMP JACKET WATER		•										

Above computed data corrected for barometer Yes/No and temperature Yes/No

• Laboratory readings.

Refer to Specification Sheet No.______

Refer to Curve Sheet No.______

Revised January 1921 by the Society of Automotive Engineers, Inc., 29 West 39th St., New York City.

S. A. E. DIESEL-ENGINE TESTING FORMS **LOG SHEET—C** **TEST No.___**

Name__________No. Cyls._____	Fuel___B.T.U. per Lb.____Sp. Grav.___at___deg. fahr.
Model__________Serial No._____	Dynamometer__________Arm (R)_____ft.
Bore_____In. Stroke_____In. Displ. (D)_____Cu. In.	Humidity__________per cent.
Laboratory__________Date_____	Lub. Oil_____Grade_____Cold Test_____deg. fahr.
Observers__________	Saybolt Univ. Vis. at 130 deg. fahr.____At 210 deg. fahr.____

BRAKE HORSEPOWER AND FUEL CONSUMPTION

Run Number	Symbol	Formula	1	2	3	4	5	6	7	8	9	10
Time Started		*										
Duration of Run—Min.	t	*										
Counter Start	C_s	*										
Counter Finish	C_f	*										
Total Rev.	r	$C_f - C_s$										
Average R.P.M.	N	$\frac{r}{t}$										
Barometer In. Mercury		*										
Room Temp.		*										
Correction Factor	C.F.	$\frac{P_s}{P_o} \times \sqrt{\frac{T_o}{T_s}}$										
Brake Load at Arm R	P	*										
Brake Load Corrected	P_c	CF×P										
Torque Lb.-Ft.	T	PR										
Brake M.E.P.	bp	$\frac{150.8\ T}{D}$										
Brake Hp.	B.Hp	$\frac{PRN}{5252}$										
Friction Hp. at N	F Hp	F.Hp Curve										
Indicated Hp.	I.Hp	B Hp.+F.Hp										
Mechanical Efficiency	M.E.	$\frac{B.Hp.}{I.Hp.}$										
Water, Gal. per Min.		*										
Temp. Jacket Water—In		*										
Temp Jacket Water—Out		*										
Temp. Oil—In		*										
Temp Oil—Out		*										
Temp. Exhaust	x	*										
Oil Press., Lb.		*										
Wt. Fuel Start	W_s	*										
Wt. Fuel Finish	W_f	*										
Lb. Fuel Used	W	$W_s - W_f$										
Lb. Fuel Per B.Hp. Hr.	F	$\frac{60\ W}{t \times B.Hp.}$										
Thermal Eff. Re B.Hp.	T.E.	$\frac{2545}{F \times B.T.U.}$										

FRICTION HP.

Run Number	Symbol	Formula	1	2	3	4	5	6	7	8	9	10
Time Started		*										
Duration of Run—Min	t	*										
Counter Start	C_s	*										
Counter Finish	C_f	*										
Average R.P.M.	n	$\frac{C_f - C_s}{t}$										
Brake Load at Arm R	p	*										
Friction Hp.[1]	F.Hp.	$\frac{pRn}{5252}$										
Mean Temp. Jacket Water		*										

Report of Diesel Engine Division adopted by the Society, January, ...

Above computed data corrected for barometer (Yes) and temperature (Yes)
*Laboratory readings (No) (No)
All temperatures are degrees fahrenheit

Refer to Specification Sheet No..
Refer to Curve Sheet No..

LOG SHEET D

S.A.E. Engine Testing Forms

Name and Model.....................................

No. Cyls..... Bore....in. Stroke.... Displ. (D)....cu. in.

S.A.E. Log Sheet C

Fuel........ B.T.U. per lb....... Sp. Grav.....at....°F.

Dynamometer............. Arm (R)...........ft.

Room temp....... °F. Bar.......in. Hg. Humidity......%

Laboratory............... Date.....................

Observers ....

Run number	Symbol	Formula	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Time started		*														
Duration of run—min.	t	*														
Counter start	C_o	*														
Counter finish	C_t	*														
Total rev.	r	$C_t - C_o$														
Av. R.P.M.	N	$\frac{r}{t}$														
Brake load at arm R	P	*														
Torque in lb.-ft.	T	P.R.														
Brake M.E.P.	ηp	$\frac{150 \cdot 8\ T.}{D}$														
Brake h.p.	B.H.P.	$\frac{PRN}{5252 \cdot 1}$														
Friction h.p. at N	F.H.P.	F.H.P. Curve														
Indicated h.p.	I.H.P.	B.H.P.+F.H.P.														

and Fuel Consumption.

Brake Horse-power	Mechanical efficiency	M.E.	$\frac{\text{B.H.P.}}{\text{I.H.P.}}$														
	Temp. jacket water—in		*														
	Temp. jacket water—out		*														
	Temp. air to carb.		*														
	Wt. fuel start	W_o	*														
	Wt. fuel finish	W_t	*														
	Lb. fuel used	W	$W_o - W_t$														
	Lb. fuel per B.H.P. hr.	F	$\frac{60W}{t \times \text{B.H.P.}}$														
	Thermal eff. re B.H.P.	T.E.	$\frac{2545}{F \times \text{B.T.U.}}$														
Friction H.P.	Time started		*														
	Duration of run—min.	t	*														
	Counter start	C_o	*														
	Counter finish	C_t	*														
	Av. R.P.M.	n	$\frac{C_t - C_o}{t}$														
	Brake load at arm R	p	*														
	Friction h.p.	F.H.P.	$\frac{pRn}{5252 \cdot 1}$														
	Mean temp. Jacket water		*														

* Laboratory Readings. See also Specification Sheet E, and Curve Sheet, Fig. 32.

LOG SHEET E

S.A.E. Engine Testing Forms—Specification Sheet—B

Name and model.................................... Date of test..............
Manufacturer..
*(1) General type.................................... Cycle..............
(2) No. of cyls........ Bore........in., Stroke........in., Piston displ. per cyl........cu. in., Total............cu. in.
(3) Compression vol. (V_c)........cu. in., Total vol. of cyl. (V)........cu. in., Compression ratio $= \frac{\text{Comp. vol.}}{\text{Total vol.}} = \frac{V_c}{V} =$ ......
Compression Pressure............lb. gauge at............r.p.m.
(4) Type of cyl. casting.................................... Matl..............
(5) Type of valves.................................... Location..............
(6) Cooling system..
(7) Piston, type.................................... Matl..............
Wt. with rings and pin........lb., Length..........in., Distance centre of pin to top of piston................in.
(8) Piston-rings, No. per piston............ Type.................... Width...............in.
(9) Connecting-rod, type..
Length, c. to c................in. Weight, upper end...........lb., Lower end...........lb., Total...........lb.
(10) Piston-rod bearings, diam..........in., Total length............in., Matl................ Location..............
(11) Connecting-rod bearings, diam............in., Length.............in., Matl........... Type..............
(12) Crankshaft bearings, No...................... Diams..............................
Material...................... Lengths..............................
(13) Camshaft bearings, No...................... Diams..............................
Material...................... Lengths..............................
(14) Type of cams...................... Type of valve-lifters..............................
(15) Inlet valves, No. per cyl............o.d...........in., Port diam.........in., Lift...........in., Seat angle....deg.
(16) Exhaust valves, No. per cyl.........o.d...........in., Port diam..........in., Lift..........in., Seat angle....deg.
(17) Weight of valve reciprocating parts, inlet...........lb., Exhaust...............lb. ..

(18) Valve-spring tension, inlet open.......... lb., Closed..........lb., Exhaust open..........lb., Closed............lb.

(19) Valve-timing, inlet valve opens..........deg...........top centre, Closes....................deg. after lower centre
Exhaust valve opens............deg. before lower centre, Closes..............deg...........top centre

(20) Flywheel, o.d...................in., Weight................lb. Moment of inertia..............................

(21) Weight of engine................lb., including..
....

Carburation

(22) Carburettor, name and model.. Nom. size............in.

(23) Specification (size of nozzles, etc.)...

(24) How heated..

(25) General principles of operation..
....

(26) Description of intake pipe..
....

Ignition

(27) Name and type of system...

(28) Type of distributor............................. Firing order.................................

(29) Type of breaker...............................Maximum spark advance..........deg., Retard........deg.

(30) Spark plugs, name and type... Size....................in.

(31) Location.. Gap....................in.

Lubrication System

(32) Type and description..
....
....

* Notes corresponding to these numbers are given under " Rules and Directions " in an Appendix.

shown on page 65; this is, of course, a reduction of the actual sheet, which measured 8 × 11 inches over all. The squares were ruled in red, but the other printed matter was in black type.

The complete record of the test should include a brief description of the object of the test, apparatus employed, any special feature, the test assistants' names, calibration results (for the measuring apparatus), and four standard sheets of results as follows: (*a*) Specification of the engine (similar to Log Sheet E); (*b*) Observers' original readings; (*c*) Computed results from (*b*), and (*d*) Plotted results from (*b*).

It is often an advantage to include a summary of conclusions, deductions, and recommendations by the assistant in charge. The most convenient method of filing is by the use of perforated test sheets, all of the test data and results being filed in the one cover. The test and log sheets can be obtained in pads with filing perforations already made.

Special and Research Tests.—Many other special tests are frequently required to be carried out in addition to those enumerated in the previous section, and in such cases special apparatus—usually of a physical research character—is required. On the other hand, it is possible to carry out some of these special tests without any elaborate equipment. It is often necessary to be able to determine the following quantities, in addition to the ones previously cited: (1) The air-to-fuel ratio for power and thermal efficiency tests. (2) Fuel consumption tests, with various degrees of throttling. (3) Temperatures of the exhaust and inlet gases, the valves, cylinder walls and pistons. (4) Variation of power and efficiencies with compression ratio change. (5) Investigation of effects of oil burning upon the other factors. (6) Detonation pressures, temperatures, compression ratios, etc. (7) Total heat losses for heat balance chart. (8) Pressures in the cylinder, inlet, and exhaust manifold. (9) Combustion measurements. (10) Supercharging and stratified charge effects, etc. Most of these quantities and their measurements are dealt with in subsequent chapters, so that a few brief remarks only are necessary here.

Mixture Range Tests.—In the first place, tests which include the variation of the mixture strength can be carried out quite readily with the needle-valve adjustment on the carburettor. Following the full power tests on the engine with the richer mixtures, the needle-valve in the carburettor jet is closed slightly and a similar set of torque, temperature, air, oil, and fuel consumption readings is taken. The tests are then repeated, with progressively weaker and weaker mixtures right down to the minimum.

After the mixture range tests the engine is again motored round with the ignition and circulating water "off," and the total losses carefully determined. During the mixture range tests the speed

must be kept constant, and the throttle full open ; it is hardly needless to add that no other variables must be introduced.

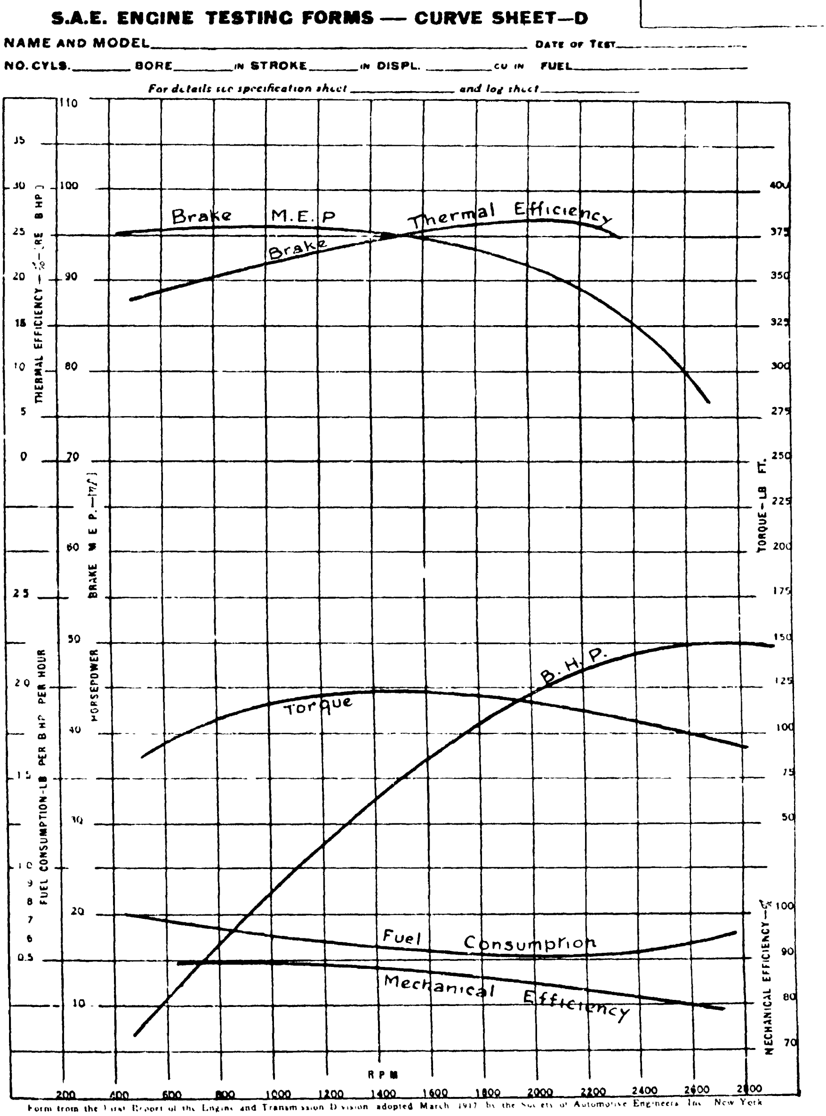


FIG. 32.—The Society of Automotive Engineers standard test chart, with test results from a typical small engine.

Next, the mixture range tests are repeated at other definite speeds, say at 1000, 1500, 2000, 2500, etc., r.p.m.

Frictional Loss Analysis.—At each speed, motoring tests should be made in order to obtain the total losses, and light-spring indicator diagrams also should be taken, in order to separate out the *pumping* from the *frictional* losses.

Finally, the frictional losses are analysed step by step in the following manner : The engine is first motored around immediately after switching off the ignition, the circulating water temperature being maintained at its mean working value ; it is necessary for certain of these tests to provide an auxiliary heater to the water supply reservoir, or to vary the rate of circulation of the water ; the power is then measured. The sparking plugs are next removed and the power required again to motor the engine around is measured. The difference between the two measured powers will represent the pumping losses, very nearly. Next, the valve gear is placed out of action (by disconnecting the drive), the engine motored around and the power measured.

The pistons and connecting rods can be removed and the power required to rotate the crankshaft and auxiliaries measured. Similarly, by removing the magneto, water pump, and other engine-driven auxiliaries, the power required to drive these parts can be determined by the same method of differences.

Very careful measurements of the total power are required, and the engine speeds and jacket water temperature should be maintained constant ; the latter is necessary for oil temperature constancy. Occasionally it is more convenient to measure the power required to drive such parts as the magneto, the water pump, electric generator, tyre pump, and similar auxiliaries, by means of a separate small, swinging field type of dynamometer.

It is usual to express the power losses thus measured in terms of the piston area, i.e. in pounds per square inch. The following values are given by Ricardo for a 100 h.p. six-cylinder petrol engine :

Bearing friction	0·75 to 1·0	lb. per sq. in. (mean pressure).
Valve gear .	0·75 to 0·80	,, ,,
Magneto . .	0·05 to 0·10	,, ,,
Oil pump .	0·15 to 0·25	,, ,,
Water pump .	0·30 to 0·50	,, ,,
Total .	2·00 to 2·65	,, ,,

The power required to drive a four-cylinder magneto is about 10 h.p. at 2000 r.p.m., whilst the centrifugal type of circulating pump absorbs about ½ h.p. at this speed.

Further information on engine power losses is given in Chapter VI.

Detonation Tests.—If the engine is of the normal fixed compression type, its detonation tendencies can be investigated firstly by measuring the maximum explosion pressures over a range of

mixture strengths, using for this purpose the 'Farnboro' maximum pressure gauge, or the Okill gauge, under given conditions of engine speed, jacket water temperature, and ignition. Secondly, by varying the type of fuel. In this case it is desirable to employ two standard qualities of fuel, namely, a light, volatile petrol freed from aromatic constituents, by a sulphonation process, and known as "Aromatic-free" petrol, and another petrol containing a known relatively large percentage of aromatics, such as toluene, benzene, and xylene. The aromatic-free petrol is suitable for low compression, and the aromatic petrol for high compression engines. The exact limits of detonation of these two standard fuels are determinable from compression variation tests on a special high thermal efficiency engine.[1]

By varying the proportions of these two fuels in any mixture, or by adding toluene to the former, the tendency of any engine to detonate can be compared with that of the standard, or variable compression engine. A careful note should be made of the compression pressures at which detonation occurs, the ignition setting jacket water temperature, and range of explosion pressures. The following table gives the properties of two standard test fuels employed by Ricardo:—

TABLE VII

Properties of Standard Test Fuels

	Aromatic-Free	Aromatic	Units
Specific gravity	0·718	0·782	
Final boiling-point † . . .	175°	175°	Centigrade
Lower calorific value (corrected) .	19,200	18,580	B.T.Us. per lb.
Mixture ratio giving complete combustion	15·05/1	14·3/1	
Total internal energy of mixture giving complete combustion .	48·5	48·15	Ft.-lb. per cu. in.
Highest useful compression-ratio in variable compression engine .	4·85/1	6·0/1	
Toluene number	0	38	

† 98 per cent. distilled over at this temperature.

It happens, occasionally, as in the case of sleeve-valve engines, that aromatic-free petrol will not detonate under any circumstances; in these cases it is necessary to add another liquid such as paraffin, or ether, in order to make it detonate.

In the case of toluene, benzene, and xylene, these fuels cannot be made to detonate at any compression up to 7·5 : 1.

[1] *Automobile Engineer*, August, 1923.

The method employed for ascertaining the octane number of a fuel can also be adapted to investigations of the detonation tendencies of petrol engines, since the proportions of the two standard fuels used, namely, *heptane* and *iso-octane* can be varied as required and the octane number of the fuel that will just cause detonation under fixed test conditions taken as a measure of the detonation tendency of the particular engine as compared with, say, a standard variable compression or C.F.R. engine.

More Recent Methods of Detecting Detonation.—Several alternative methods have been employed to detect detonation in

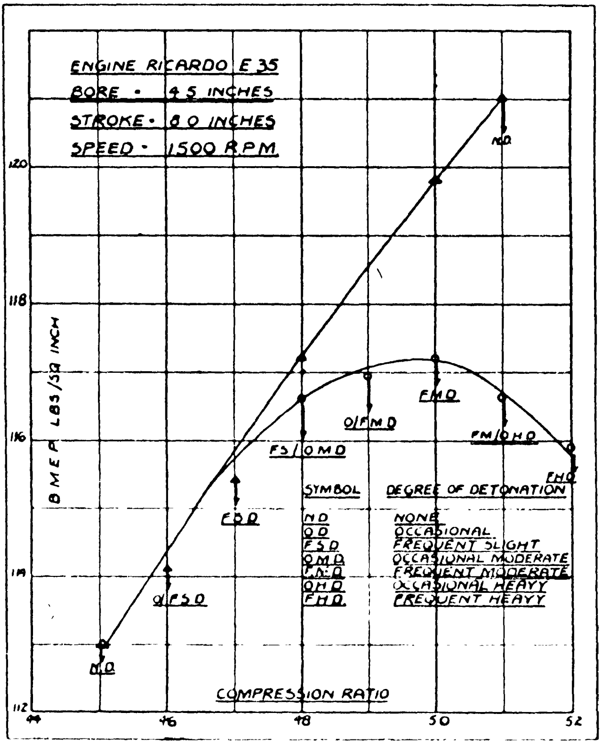


FIG. 33.—Effect of detonation on power output.

petrol engines under test, and in this connection some of these methods have been put forward as suggested methods of determining the knock ratings of fuels.

If the power output of an engine be measured, by means of a dynamometer at full throttle, it will be found that as the compression ratio is raised the power falls off after a certain degree of detonation is attained ; this is shown in Fig. 33 and affords the basis of a method of detecting detonation. The upper curve represents the B.M.E.P. values obtained from a fuel in which detonation was

absent at all of the compression ratios, up to 5·1 : 1 used during the tests.

Indicator diagrams taken from petrol engines operating under detonation conditions show pronounced variations in the pressures during the pressure rise (immediately after ignition) period and early stages of the expansion stroke as shown in Fig. 34.

The occurrence of higher maximum pressure values than for smooth running conditions is also a feature of these indicator diagrams.

Measurements of cylinder head temperatures, heat losses and rates of pressure rise also afford satisfactory means of detecting detonation. Mention should also be made of high-speed cinematograph pictures which have been taken through quartz windows in the cylinder wall during the combustion stages in actual engines by Withrow and Rassweiler [1] which prove that when detonation occurs the rate of combustion—as shown by the intensely white

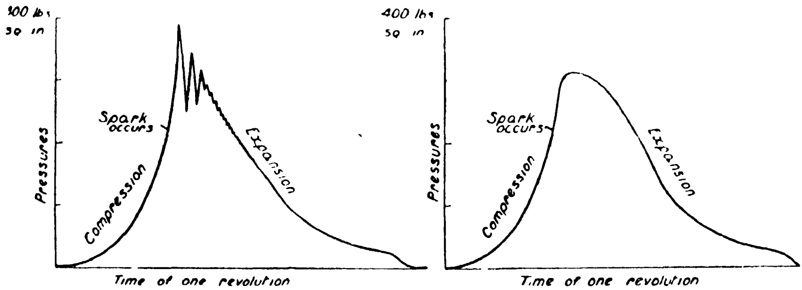


FIG. 34.—Detonation effect shown on indicator diagrams. (Left) Detonating conditions. (Right) The same engine running on a non-detonating grade of fuel.

areas of the photographed cylinder head—is accelerated considerably as compared with the normal process of non-detonating combustion.

Colour photographs have been obtained from combustion chambers during actual operation of engines by the Ethyl Gasoline Co. Laboratories, New York. A characteristic feature of these photographs is the carbon-yellow region which appears under detonation conditions, whereas under non-detonation ones the same regions appear as light blue to blue areas.

The phenomenon of detonation has also been studied chemically by several investigators using methods based upon the chemical analyses of the gases during the later stages of combustion. By the use of special apparatus it is possible to sample the gases at different parts of the combustion chamber ; moreover, these samples can be taken at different crank-angle positions before and after top dead-centre. From the results of such analyses the proportions of oxygen, carbon monoxide, and carbon dioxide can be deter-

[1] *Jour. Soc. Automotive Engrs., U.S.A.*, Aug. 1936.

mined. The oxygen contents, if taken at different times or crank-angle periods, are a measure of the rate of combustion, for the rate of diminution in the proportion of oxygen is an indication of the combustion rate. Under normal non-detonating conditions there appears to be a sharp but narrow zone of combustion which proceeds from the ignition centre with a velocity dependent upon the engine speed, mixture strength, etc. It has been found that behind this zone no oxygen can be detected so that it is concluded that combustion is complete in this region. The actual velocity of the combustion zone is greater in the interior of the combustion chamber than in the vicinity of the walls.

When the engine is operating under detonating conditions it is found that a greater velocity occurs for the outwardly travelling combustion zone after the latter has traversed about three-quarters of its total movement ; thereafter a greatly accelerated rate occurs.

Another chemical method is based upon observation of the presence and amounts of aldehydes and peroxides when detonation occurs.

A practical method of detecting and measuring detonation, devised by Midgley and Boyd, is that known as the "bouncing pin" one ; this method has been widely used and is the basis of the Co-operative Research Fuel or C.F.R. engine.[1] test method used for assessing the anti-knock qualities of various fuels. The "bouncing pin" device is screwed into the wall of the cylinder in the combustion chamber end and is arranged to form what is essentially a small thin-walled section of the wall. One end of the bouncing pin rests on this wall. The pin is guided so that its movements occur at right angles to the wall, the motion in this direction being controlled by a spring which when deflected closes an electrical circuit. When thus closed a water electrolysis apparatus is brought into operation. The amount of detonation is then measured by collecting the gas evolved through the passage of the current through a 10 per cent. sulphuric acid solution. The quantity of gas evolved per unit time thus serves as a measure of the detonation period.

Fig. 35 shows, diagrammatically, the Midgley bouncing pin device as used for detonation tests. It should be mentioned that visual observations of detonating conditions can be arranged for, by means of an electric relay switch device which is arranged to complete an electric circuit when detonation occurs and thus to cause a lamp to light. The bouncing pin method, whilst extremely useful for comparing pronounced detonations in the case of different fuels is not altogether satisfactory for detecting incipient detonation—a phenomenon which requires rather more delicate means of indication. Further, it is not altogether satisfactory for high octane fuel tests.

[1] See Appendix No. II.

The results of some research work by N. MacCoull [1] are given in Fig. 36 for fuels which would detonate under the operating conditions and also a fuel that would not detonate over the range of compressions, namely, 4 : 1 to 7 : 1 employed. The temperatures of the sparking plug, piston head, exhaust valve, intake, exhaust gases and a U.S. Army "hot-plug" were measured and the results are here reproduced in the form of the graphs given in Fig. 36.

In connection with these curves the letters refer to the following test conditions :—

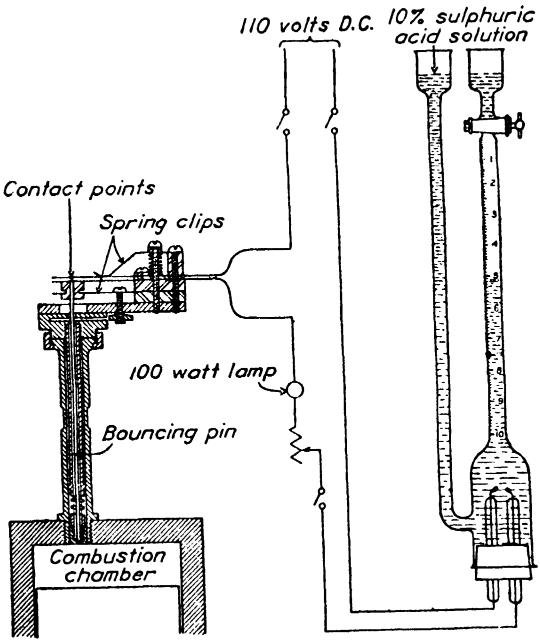


FIG. 35.—Principle of Midgley bouncing pin apparatus.

A. Results obtained with a high octane fuel which would not detonate, the ignition advance being adjusted for maximum power at each compression ratio.

B. Results of tests with a fuel which detonated audibly for a compression-ratio of 4·5 : 1, and therefore at all higher compression ratios. The ignition advance was adjusted for the compression ratios similar to A.

C. Using the same fuel as for B but with the ignition timing adjusted for maximum power at each ratio.

D. Using the same fuel as for B but with ignition timing retarded to the threshold of detonation at each compression.

[1] "Power Loss Accompanying Detonation," N. MacCoull, *Jour. Soc. Automotive Engrs.*, 1939.

The general conclusions drawn from the complete series of tests are summarized as follows:—

(1) The compression ratio at which a fuel produces border-line knock, with ignition timing for maximum power and mixture ratio for maximum knock, is called the " critical compression ratio " for the fuel used, in the engine tested.

(2) The ignition timing to be used for compression ratios above critical should be retarded from that which would give maximum power for a fuel which does not knock at these compression ratios.

(3) It appears feasible to operate an engine above the critical compression ratio provided the ignition timing is adjusted for maximum power with the fuel used. This will be a smaller advance than with a fuel which would have allowed a higher critical compression. For each increase of compression ratio there was an increase in power output, until a limit was set by pre-ignition.

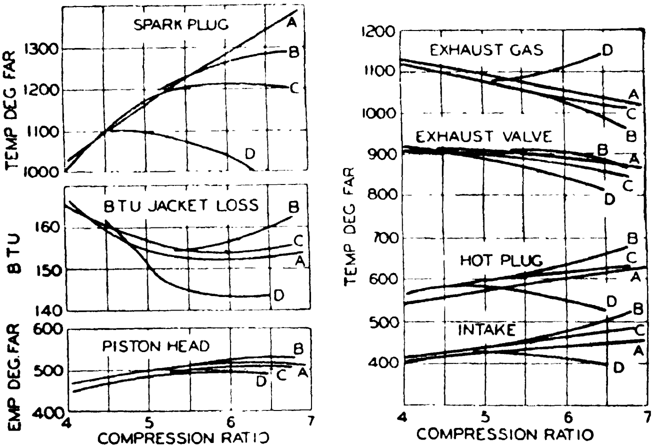


FIG. 36.—Engine temperatures and jacket losses under detonating conditions.

(4) The tendency for a fuel to pre-ignite varied inversely with its anti-knock value, but was profoundly influenced by the design of the cylinder head, which influenced the degree of cooling, as well as the thermal properties of the spark plugs. It was only slightly influenced by spark advance.

(5) It appears desirable to use a compression at least one ratio higher than critical for the fuel to be used, and retard the spark sufficiently to eliminate knock. This results in higher engine efficiency; what is still more important, if it becomes desirable to use a higher octane fuel, it makes possible a noticeable gain in power by suitably advancing the spark.

Other Practical Effects of Detonation.—Apart from the characteristic knocking sound produced by detonating conditions, the power output, as mentioned previously, is reduced. This is

shown by the results of tests made with the Ricardo E. 35 variable compression engine which clearly indicated that detonation is associated with *an increase in the heat losses* to the cylinder walls and pistons and *a loss of thermal efficiency*.

Detonation is also accompanied by an increase in the maximum cylinder pressure ; in some instances the measured values of these cylinder pressures have been as much as twice the normal values. The extremely rapid combustion process which is known to be associated with detonation, and the high " impact " pressure effects have been known to cause fracture of the pistons of engines, whilst the increased heat losses to the pistons and valves have resulted in burning the top lands of the former and the heads of the valves themselves.

Further, it has been shown that the increased cylinder temperatures have, in severe cases of detonation, resulted in the piston crowns collapsing in the centre and sometimes in burning of the cylinder head. Other detrimental results of prolonged running under detonating conditions include gumming of the piston rings in their grooves, the loosening of valve inserts in the cylinder heads, and burning of the sides of the piston due to blow-by of the hot gases. If there is any appreciable carbon deposit on the combustion chamber walls, valve tops and piston crown before detonation occurs, it is generally found that the violent pressure conditions which subsequently result from detonation dislodge this carbon, whence it is blown out through the exhaust pipe. An instance is given by G. D. Boerlage of a Ford engine, the cylinder head of which had a hard carbon deposit about 2 mm. thick. This carbon was completely cleared away by running the engine for about ten minutes upon a low grade of petrol which promoted detonation.

When aluminium alloy pistons and cylinder heads are employed in petrol engines the effect of pronounced detonation is not only to eliminate all carbon deposits, but, in some instances, to cause a rough or pitted appearance of the metal.

The effects described are associated with pronounced degrees or long periods of detonation. A small degree of detonation may be experienced in aircraft engines, without loss of power or detrimental results to the engines themselves, but it has been established that even small detonation tendencies result in a redistribution of the heat losses, whereby less goes to the exhaust gases and more to the cylinder walls and pistons, thus causing increased temperatures with a tendency to cause valve burning and piston ring gumming.

When an engine is running under detonating conditions on maximum power mixture strength, i.e. about 15 per cent. rich, the exhaust flame tends to become more yellow, whereas without detonation it is of a bluish colour with occasional splashes of yellow and orange.

Puffs of black exhaust smoke also indicate detonation; these occur occasionally and are distinct from the black smoke of very rich mixtures.

Notes on Detonation.—In examining any given engine for the causes of detonation, the following principles, which have been established as a result of research [1] may be summarized as follows: (1) Detonation depends upon the form of the combustion chamber. (2) Detonation depends upon the number and position of the sparking plugs; it is a minimum the greater the number of the plugs and the more equal their spacing. (3) The inlet temperature, and to some extent, also, the cylinder temperature influence detonation; the lower the inlet temperature the greater the allowable compression pressure without detonation occurring. (4) Detonation depends also upon the degree of turbulence, the tendency to detonate increasing with decreased turbulence. (5) Detonation is most apparent over the range of mixture which lies between the points of maximum economy and maximum power, so that it cannot, in practice, be avoided by changing the mixture strength.

Detonation and Mixture Strength.—The importance of making comparative tests at similar mixture strengths is emphasized by the fact that higher compression ratios can be employed, before detonation begins, with mixtures which are both richer and weaker than the one giving complete combustion. In this connection the results of some tests made by Ricardo on a variable compression engine running at full throttle and under constant speed and temperature conditions may be quoted. In each case the compression ratio was adjusted to the particular mixture strength until detonation was just apparent. The results are shown in the following table:—

TABLE VIIA

Compression Ratio and Mixture Strength

	Weak			Rich				
Mixture strength .	20%	10%	Correct	10%	20%	30%	40%	50%
Compression ratio just causing detonation	5·04	4·89	4·85	4·85	4·88	4·95	5·05	5·22

Detonation and Exhaust Gas Dilution.—If the mixture fed to the engine from the carburettor is diluted with exhaust gases from the same engine, the compression pressure can be raised appreciably before detonation, previously in evidence, occurs again. The

[1] Aeronautical Research Committee Report No. 57, "Detonation in Internal Combustion Engines."

results of some tests made by Ricardo are reproduced in Fig. 37, which shows how the compression ratio was increased from the detonating value of 4·85 : 1 by increased amounts of exhaust gas; in each case the compression was raised until detonation was just apparent. Parallel tests with other inert gases such as nitrogen and CO_2 gave similar results. It was pointed out that the effectiveness of these inert gases appeared to be related to their specific heats and thus to their direct influence upon flame temperature. In regard to any detrimental effect of turbulence upon detonation, this does not appear to be an important factor, according to Ricardo, who actually found that with combustion chambers designed to

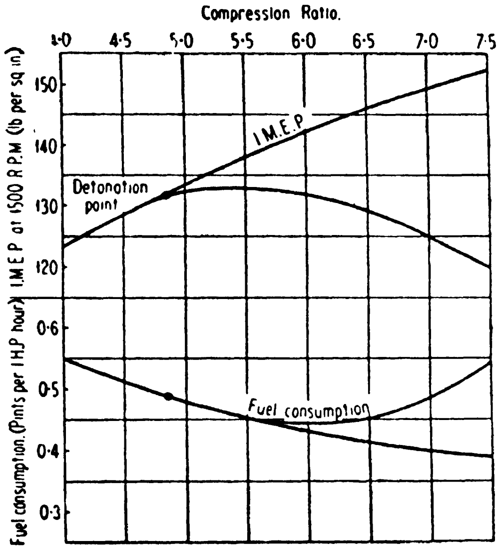


FIG. 37.—Increasing the compression pressure, without detonation, by means of the exhaust gases. The upper I.M.E.P. curve refers to non-detonating and the lower ones to detonating conditions.

give a high degree of turbulence there was a marked reduction in the tendency to detonate.

It has been possible, only, to give a brief outline of the subject of detonation chiefly from the practical point of view. For fuller information on the theoretical and certain other practical aspects of the subject the reader is referred to the account given in the chapter on "The Combustion Process" in the author's book on "Aircraft Engines," Vol. I (Chapman & Hall, Ltd.).

General Considerations. 1. **Application of Test Results.** —The engine manufacturer and the practical engineer will ask how the results of engine tests such as those described can be applied directly to commercial engines. Evidently it is one thing to take an engine and substitute another special carburettor, and

variable water cooling device, to use special fuels, carburettor heating means, ignition devices, valve timings, etc., in order to obtain the best results, and another to apply the experience gained to practice.

The preliminary "tuning-up" tests will indicate the best ignition timing position and valve clearances, and this information can be applied directly.

The searching tests for volumetric efficiency, thermal efficiency, and B.H.P. output at different speeds, will enable the test engineer to say at once, by comparison with the standard values available, whether the results are satisfactory, and if not, wherein lie the causes. It may so happen that the induction system is badly designed, the valve area or opening period inadequate; this will be revealed by the volumetric efficiency and also the normal and maximum power tests. Or, again, the fuel consumption per B.H.P. hour may be abnormal; the design of the combustion chamber and the engine losses will be examined for a possible explanation, and so on.

In all these considerations it is understood, of course, that the mixture strength, ignition timing, the cooling water temperature, are maintained at their most favourable values, in order to elucidate the causes of inefficiency.

It will, therefore, be evident that the performance of any engine can be accurately gauged, and definite conclusions arrived at by the test engineer as to the merits or demerits of the engine.

At the end of Chapter I will be found a useful account of standard test results and performance standards with which the results of tests upon any new type of engine, no matter what its design, or working cycle, can be compared. Having mentioned these facts, we will pass on to a brief consideration of the practical application of the test data, in the case of any engine which has been tested in the manner prescribed under the heading "Tests of New Type Engines."

In the first place, *the test carburettor is replaced* by the standard one, or, if the former is of the standard type, the needle-valve is removed from the jet, and different fixed-size jets are fitted, until the result of a brief series of B.H.P. and fuel and air consumption tests indicate that the jet size gives the best compromise for power and fuel economy in comparison with the variable jet results.

Also, the artificial method of *heating the carburettor*, e.g., the electrical resistance one described, is replaced with the jacket cooling water or exhaust heated means normally provided. Any falling off in power or efficiency, with normal engine acceleration, can often be remedied by a suitable alteration in the carburettor heating system or thermostat regulation. Measurements of inlet pipe mixture temperature will be found useful in this respect.

Next, the normal *cooling water circulating* (i.e. pump or thermo-syphon) *arrangement* is substituted for the test system, and a few sets of measurements of water temperatures made under working conditions. Here, again, suitable alterations can usually be made, in order to simulate the best test conditions.

It is, of course, frequently necessary to carry out some of the final tests, more particularly the acceleration and jacket cooling water temperature *tests upon the road itself.*

One most useful result which the laboratory or bench tests will reveal, is whether the compression ratio of the engine is the most suitable one from the point of view of maximum power, and detonation. It is often the case that the results will indicate a beneficial *change of compression ratio,* and in this respect the manufacturer is able by substituting another design of piston, namely, one with a different distance between the gudgeon pin axis and the crown, by altering the connecting rod length, by re-designing the detachable cylinder head, or even by introducing a metal liner between the cylinder holding down flange and the crankcase, to make the necessary alteration for test purposes.

2. **Calibration of Apparatus.**—It is necessary that all apparatus employed for engine tests be initially and periodically checked for accuracy, more particularly in cases where non-technical or semi-skilled assistants have to carry out routine and other tests. Many hours of labour and much money may be wasted in this respect, whilst the test results are not only erroneous, but misleading. In the case of motor manufacturing works, and experimental workshops, the test apparatus is apt, partly by long use and partly by oversight, to get out of adjustment and become inaccurate. If no means are available for ascertaining the veracity of their readings, they should be checked by a representative of the instrument manufacturers, or returned to the manufacturers themselves for calibration.

Certain testing institutions, such as The National Physical Laboratory, Teddington, Middlesex, and Faraday House, Southampton Row, London, as well as several commercial testing institutions and universities and colleges, are now available for the purposes of checking test apparatus, and issuing a certificate of performance and calibration, for a nominal fee. On the other hand, the technically trained assistant should be able to check the accuracy of all of his test apparatus.

The well-trained test engineer will make a rule of checking his test results, by means of alternative tests, often utilizing different methods and apparatus.

To take one example, the mixture strength values obtained by separate fuel and air measurements can be checked by analysing the exhaust gases, a procedure which should always be followed in important tests.

Period of Tests.—The ordinary type of bench test can usually be carried out expeditiously by two trained assistants. It is not always possible to ensure uniform behaviour of a new type of engine, even after it has been tuned-up and given a preliminary run, but in normal cases it is possible for two assistants to carry out a bench test of fuel consumption B.H.P. and engine losses, over the whole range of mixture strengths, in about an hour.

The test plant must, of course, be arranged conveniently, so that all of the controls are centralized, and the instruments and scales also grouped together.

With practice one assistant can look after the engine controls, namely, the throttle and ignition levers, the dynamometer load (if electrical), the carburettor jet control, and the water temperature and quantity control, and can actually take fuel readings whilst the other man reads the dynamometer torque and takes speed readings.

With three assistants, one man can confine his attention to the controls only, whilst another man looks after the fuel readings and circulating water, the third one taking power and speed readings.

For special tests necessitating air and temperature measurements one senior and three other test assistants are required as a minimum. The senior man directs the tests, and, by a pre-arranged signal such as a warning alarm, or a nod of the head, starts and stops each of the tests. The additional man takes simultaneous readings of air consumption and temperature.

It is not always necessary or advisable to carry out all of the tests simultaneously, since the influences of the different factors can be determined separately. For example, having calibrated the carburettor jet-needle positions, the I.H.P. and engine losses can be determined separately, whilst volumetric efficiency and fuel consumption tests can also be carried out as separate items.

The ideal arrangement, however, would be to make simultaneous measurements of all of the quantities concerned ; this would eliminate the possibility of any individual variations vitiating the results. A party of four test assistants, properly trained and drilled, should be able to carry out a test extending over the whole mixture range up and down, together with air and water quantity (and temperature measurements), brake horse-power, and motoring tests in about an hour.

The complete series of tests necessary for arriving at definite conclusions in connection with new types of engine, including the preliminary running-in, the tuning-up, and calibration tests will, as a rule, occupy from 15 to 25 hours. Special duration tests of a tuned engine will, of course, occupy an additional length of time, depending upon the nature and object of the tests. Thus some engineers will be content with a 25 or 50 hour duration test, whilst in the case of "revolutionary" designs and new types, reliability

tests may extend up to 500 hours or even more. It is only in this manner that the effects of wear and the maintenance of normal power can be examined.

Endurance and Special Tests.—When the Knight sleeve-valve engine was submitted for test to the R.A.C. by Messrs. the Daimler Motor Co., in 1909, at the time of its adoption as standard, it was given a bench test of 132 hours (continuous running), followed by a track test at Brooklands of about 45½ hours, after which another bench test of 5¼ hours was given, making 182¾ hours, exclusive of about 10 hours spent on the road.

Aircraft engines of new design are frequently given endurance and wear tests, at normal power, of from 180 to 250 hours, without any adjustments being made, whilst new types of automobile engines are given a minimum period of continuous running at normal power of 100 hours. It is usual to specify an endurance test of this nature to be followed by a maximum power test of from 1 to 5 hours in the case of a new engine.

Air Ministry Test Requirements.—The Air Ministry official test procedure,[1] specifies an endurance test, in the case of new designs of engines, of 50 hours' duration at normal speed, and at 90 per cent. of the full power. This endurance test is made up of five non-stop runs of 10 hours' duration. Before the last hour of the endurance test the load is increased, so that the engine is running at full power at normal r.p.m. until the completion of the test. The engine must also run for at least 20 hours with a thrust applied to the airscrew shaft; this test may be carried out during the 50 hour one. In addition to these tests, the following slow running and acceleration test is made: At the conclusion of the endurance and final tests, while the engine is still warm, observations are made to ascertain that the engine runs slowly, and also speeds up to the normal r.p.m. within five seconds of opening the throttle without excessive "popping" or intermittency; the engine is also expected to run at about 50 per cent. reduction of speed from the normal r.p.m when the brake has been adjusted for 90 per cent. power at normal r.p.m.

Further, the engine is given a high speed test, being required to run for 1 hour continuously at 5 per cent. excess speed, and under load conditions optional to the manufacturer. The engine is also run for 1 hour continuously at the established maximum permissible speed and at full throttle.

Summarizing the tests referred to, and in their respective order, these comprise: (1) An endurance test of 50 hours. (2) A thrust test of 20 hours' minimum. (3) A slow-running and acceleration

[1] Air Publication 840 (Feb. 1922), with Amendments to provide for engines not exceeding 1500 c.c. (Nov. 1923). H.M. Stationery Office, Kingsway, London, W.C. 2., and later additions and amendments.

test of 30 minutes. (4) A high speed test of 1 hour. (5) A high power test.

A power curve test is given before (1) and again after (5). The engine is then dismantled and inspected for defects and wear. Finally, the engine is re-assembled and given a test of 30 minutes' duration under the same conditions as (1).

Petrol and Oil Consumption.—It is further specified that the consumption of petrol must not exceed 0·56 pint per B.H.P. hour for water-cooled engines, and 0·60 pint per B.H.P. hour for air-cooled ones; the consumption figures are taken as the average values obtained on each period of the 50 hour endurance test.

The consumption of oil for water-cooled engines must not exceed 0·025 pint per B.H.P. per hour; for air-cooled engines this fignre is 0·045.

Water Pump Delivery and Water Temperature.—It is stipulated that the pump must deliver at least 15 gallons of water per minute per 100 B.H.P. (normal), against a circuit resistance or head equivalent to 2 lb. square inch in excess of that required to overcome the hydraulic resistance of the engine, while the water temperature at the pump inlet branch should be not less than 80° C.

General Note.—The above-mentioned tests are fairly stringent ones, but most modern engines are capable of satisfying the conditions laid down.

The limitation of the endurance test for new designs to a period of 50 hours, and of 2 hours only in the case of acceptance tests for new engines of approved design, is considered to be a weak point in the above specification. A new design of engine should be given an endurance test of at least 100 hours running, and preferably of 150 hours, since no appreciable wear occurs in well-designed engines under these periods, and certain types of defect do not become manifest. New designs of automobile engines frequently run for periods equivalent to 250 hours and above.

3. **Duration of Test Runs, Etc.**—In the previous section we have considered total running periods as distinct from those of individual test runs.

The brake horse-power test runs should each extend over a minimum period of 3 minutes, and that of friction horse-power of not less than 1 minute.

Fuel consumption runs, with specially calibrated apparatus, such as the Ricardo type, average about 1 minute per test. Any shorter period leads to higher percentage timing errors, whilst longer periods allow opportunities for variations in speed and torque. The American Society of Automotive Engineers specify a minimum period of five minutes for fuel consumption tests, under constant conditions of speed, power, and temperature. They also recommend that measurements of power and fuel consumption should be

made at the lowest speed the engine is capable of running at uniformly, and at intervals of 200 r.p.m. upwards. The maximum permissible speed variation is 50 r.p.m. The mean circulating water temperature, they state, should be kept at 175° F. for comparative test purposes, and should not vary by more than 5° F. above or below this value during a test.

4. **Air Draught on Engine.**—Most automobile, and all aircraft engines operate under conditions which provide a relative air speed, and for this reason it is preferable to provide an equivalent quantity of air at a chosen mean velocity, blowing on to the engine.

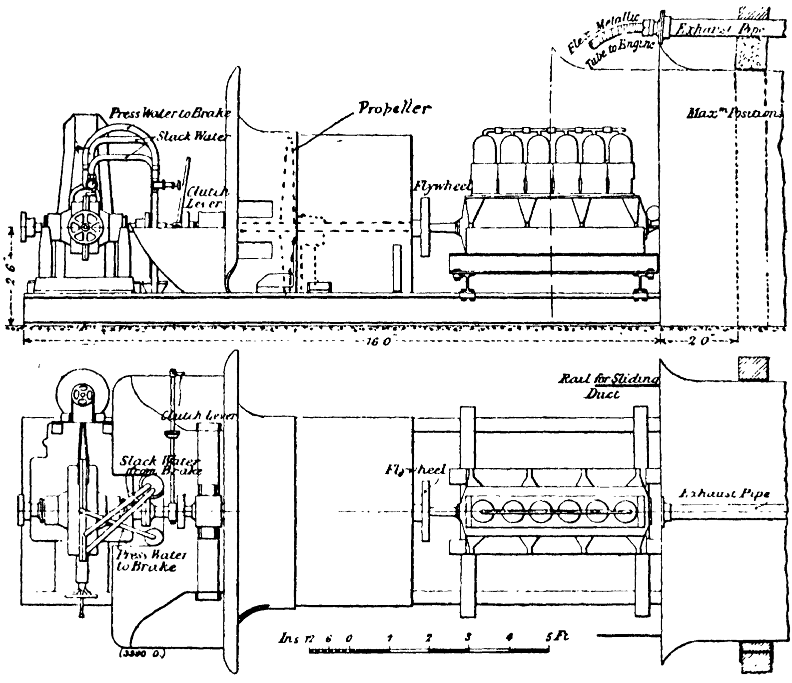


FIG. 38.—Illustrating cooling draught arrangements used in aircraft engine bench tests.

This may conveniently be carried out by affixing a fan in its designed position, in the case of automobile engines, and driving it off the crankshaft or camshaft. Alternatively a separate mechanically or electrically-driven fan (or fans) can be arranged so as to simulate actual conditions. In the case of aircraft engines, a propeller, co-axial with the crankshaft, is usually arranged, so that an equivalent air speed to that of the aircraft is given. For air-cooled engine tests, this cooling air draught is, of course, essential.

The quantity of air for an automobile should correspond to that when the engine is moving on the road at the speed corresponding

to the engine speed. The mean speed of the air during a bench test can be measured by means of pitot tubes, and in this respect it is advantageous to direct the air on to the cylinders through a metal pipe or duct in which the pitot head is placed.

In assigning the air speed, account should be taken of the fact that the air velocity through the radiator core is only about 50 to 70 per cent. of the road speed.

By driving the cooling fan off some part of the engine, the amount of air delivered will vary (as is required) with the engine speed.

Further reference to this subject is given on pages 179, 377 and 380.

Road Tests of Automobile Engines.—In most instances after a new engine has been fitted to its chassis the complete chassis unit is given a road test by an expert tester in order to discover any weak points in the engine and transmission, to locate the sources of any noises that may occur and to check the tunings of the ignition system and carburettor, valve clearances, etc.

The tests include acceleration, maximum speed (on special tracks) and hill-climbing, and the distance run varies from 25 to 100 miles according to the grade of car. Entirely new types of engine and/or transmission may be given the equivalent of a season's ordinary running period covering many thousands of miles. An expert tester is able to ascertain all of the practical information required; to express a reliable opinion concerning the merits or otherwise of a new engine and to advise upon detail improvements in regard to accessibility and adjustments, etc.

Road testing involves driving under conditions of frequently varying engine speeds and loads and with the engine subjected to road shocks. The general procedure adopted for road tests may be illustrated by a brief account of the method adopted by a leading British car manufacturer, namely, as follows :—

The engine having passed its final bench tests for maximum power output, oil and fuel consumption, and for endurance over a limited period, is mounted on the chassis of the test car and the whole unit is then given a chassis test on a special dynamometer. The chassis is then fitted with a set of mudguards and valances, bonnet, dashboard (with instruments), and the driver's seat. Usually, no body is fitted, but the chassis is loaded with pig-iron bar to about the same weight as the production car, with full complement of passengers.

Three separate road tests are made, each covering a distance of 60 miles. The first test is carried out with the object of discovering any inherent faults and for the purpose of any necessary adjustments. The second test is made for ascertaining whether the faults revealed by the first test have satisfactorily been remedied and the adjustments have effected the desired results. The final test is then made with the actual car body in place in order to test for springing, clearances, rattle, and other noise effects.

During the first two road tests the fuel consumption is measured by means of an auxiliary tank of one gallon capacity, noting the speedometer readings at the commencement and completion of draining the tank; this enables the mileage per gallon of fuel consumed to be computed. The tester is supplied with a special printed form having column headings and blank spaces to be filled in after the first and final tests. The car type and engine and chassis numbers are entered at the head of the form and the printed columns (which number over forty, arranged horizontally) are sub-divided into headings which include the engine, clutch, gears, brakes, wheels, springs, back-axle, universal joints, radiator and bodywork. Each sub-heading has a list of questions to which an answer must be given. For example, under "The Engine" are the questions: "Does it pull well?", "Are tappets noisy?", "Is water-pump all right?", "Does engine smoke?", "Has carburettor been adjusted for slow-running?", etc.

There are additional columns for the tester's remarks on the condition of the roads, petrol consumption, distance run, date of test, and names of tester and work's inspector. Thus, by making the answering of this questionnaire compulsory most of the essential information is obtained. In connection with the road testing of commercial vehicle engines and transmission components the Associated Equipment Company employs a special chassis fitted with a complete set of instruments for fuel, acceleration, speed, and other essential measurements. The chassis (Fig. 39) is provided with a special design of enclosed cab in which the measuring instruments are mounted. When it is desired to test any particular engine, carburettor, fuel-injection system, or other component, the vehicle is driven over a standard route near London, and a series of stopping places are arranged to simulate actual conditions of motor bus driving. The test route is 12½ miles long and the road speeds vary from 15 to 20 m.h.p. according to the particular section of the route. Stops are made for 15 seconds along the route, at distances of about one-third mile apart. During these tests readings are taken of the speed, fuel consumption, air and water temperatures, etc. In this way scientific tests can be carried out under practically identical conditions to those of the motor omnibus.

In order to obtain full information on engines, their components, and other chassis members that have passed the ordinary test stage or are actually on the market, the A.E.C. arranges to equip a number of otherwise standard omnibuses with the new part in question, and to run these on one or other of their ordinary London omnibus routes, on which a similar number of standard omnibuses (not fitted with the device to be tested) operate under identical conditions.

Periodical reports are made on the behaviour or performance

of the two sets of vehicles, and after a certain length of service, the test reports are compared or analysed at the head-quarters of the experimental department. The practical value of any new component can then be gauged under identical conditions to those under which it will, if adopted, have to work.

Referring to Fig. 40, when it is desired to carry out a test run the top tank B is filled from the lower one J by means of air pressure until the gauge glass D, shown on the front, indicates that it is full to a definite known quantity. This tank, in turn, feeds the graduated measures, which are used for taking readings between the individual sections of the route. The bottom of the graduated measures is provided with a change-over cock arranged in such a manner that the right-hand measure will be used for the first section, the left-hand for the second, and the right-hand for the third, and so on. The readings obtained for the separate sections of route are then added together and checked against the total quantity of fuel required to refill the bulk tank B, shown in the right-hand corner. By this means a double check is obtained.

The test routes which are used are divided into a number of sections having different characteristics of gradient, etc.; and it is possible to determine under which conditions fuel economy is good, or the reverse, by the taking of separate readings over these sections. For instance, when comparing one carburettor against another, it is found that whereas in the main test Instrument 1 is better than Instrument 2, under certain conditions the latter will show an improved result over the former.

The gear indicator A in the top left-hand corner is primarily to indicate the gear which the driver is using at any particular time but is also used in connection with certain tests of a special nature, e.g. the obtaining of data concerning the number of gear changes per mile on certain bus routes, etc. The throttle and ignition position indicators F and G are for the purpose of showing the settings of the corresponding controls.

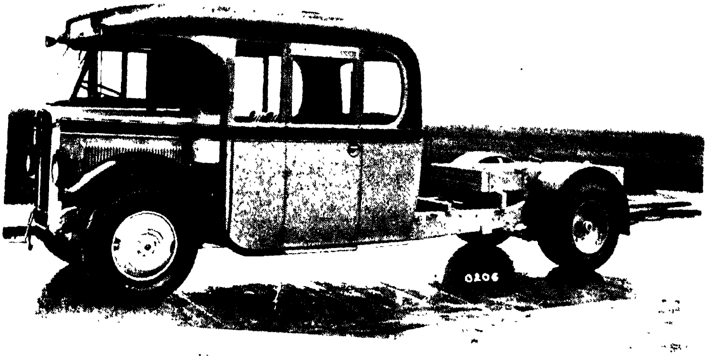


FIG. 39 The A.E.C. test vehicle

[*See page* 83.

FIG. 40 Interior of A.E.C. test vehicle, showing testing instruments, etc.

KEY. A. Gear indicator. B. Bulk tank containing measured quantity of fuel. C. Filter. D. Graduated measures for intermediate readings. E. Thermometer. F and G. Throttle and ignition indicators. H. Speedometers. J. Main storage tank.

[*To face page* 84.

Chapter III

FUEL TESTS AND EXHAUST GAS ANALYSIS

In the following considerations liquid fuels only will be dealt with, although, as will be shown, the methods are applicable, with simple modifications, to gaseous fuels.

The measurement of all the properties of a given fuel is a long and tedious matter, which the physicist and technological chemist alone are capable of dealing with, and upon the subject of which comprehensive treatises have been written and are available. It is therefore proposed to mention here only those properties which concern the ordinary experimental engineer and test assistant, namely :—

1. The Specific Gravity.
2. The Calorific Value.
3. The Amount of Fuel Used.

Specific Gravity.—The specific gravity of a liquid is the ratio of the weight of a given volume of the liquid to the weight of an equal volume of water, at the same temperature.

The determination of the specific gravity is a straightforward matter, provided that the temperature of the fuel is known accurately. It is only necessary to obtain the weight of a known volume of the liquid in order to deduce its specific gravity.

A convenient method is to weigh a body, heavier than the liquid, first in air, and then in the liquid, as shown in Fig. 41.

If W = weight of body in air, and W_1 = its weight in the liquid ; then if V = volume of body (measured),

$$\text{Specific gravity} = \frac{W - W_1}{V}.$$

The liquid should not be absorbed by the body selected for this test. For petrol, a light metallic cube is quite satisfactory.

Another method is to take a glass vessel, provided with a fairly narrow neck, and to weigh it empty ; call this weight W. Its weights when filled to exactly the same mark on the neck with distilled water and then with the fuel are next measured ; call these W_1 and W_2 respectively, then we have

$$\text{Specific gravity of fuel} = \frac{W_2 - W}{W_1 - W}.$$

It is necessary to dry the flask thoroughly between each filling and to maintain the water and the fuel at the same (known) temperatures. A small correction is also necessary for the weight of the air displaced by the liquids.

Thus, if w = weight of air (i.e. flask volume × air density), then specific gravity of fuel $= \dfrac{W_2 - W + w}{W_1 - W + w}$.

The specific gravities of the lighter internal combustion engine fuels such as petrol, benzole, paraffin, and alcohol, lie between 0·60 and 0·85. Emphasis has been laid upon the fact that it is important to know the temperature corresponding to the specific gravity.

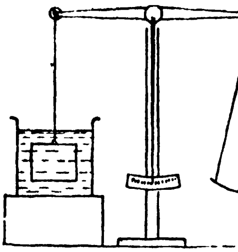


FIG. 41.

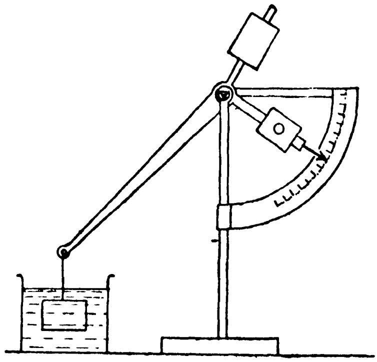


FIG. 42.—Illustrating alternative method of finding the S.G. by a single reading type balance.

This is necessary because the latter varies considerably with change of temperature, as the following results will show :—

TABLE VIII

Relation Between Specific Gravity and Temperature of Petrol (Watson)

Temperature . . .	5° C.	15° C.	25° C.
Specific gravity . . .	·730	·720	·712

A very close relation between the specific gravity and the temperature is given by

$$d_t = d_{15}\{1 - \alpha(t - 15)\}$$

where d_t and d_{15} are the densities at t° C. and 15° C. respectively, and α = mean coefficient of expansion (= ·00121 in the above example).

It is useful to remember that the weight of a gallon of fuel is equal to 10 times its specific gravity

or $$W = 10 \times \text{sp. gr.}$$

Thus a gallon of petrol of sp. gr. = 0·720 will weigh 7·2 lb.

Calorific Value of a Fuel.—The calorific value of a fuel is the amount of heat evolved, or liberated, during the combustion of unit weight of the fuel with air or oxygen.[1] In this country, the calorific value is generally expressed in British Thermal Units per pound weight of fuel.

In order to determine this value, it is necessary to burn the vapour of the fuel with air, or oxygen, in the proper proportions for correct combustion, and to measure the quantity of heat liberated.

Instruments for measuring the heat combustion of fuels (whether solid, liquid, or gaseous) are termed Calorimeters. Amongst the better-known calorimeters may be mentioned the Mahler-Donkin or "Mahler-Cook" bomb type, the Rosenhain and Roland-Wild types for solid fuels, whilst for gaseous fuels, such as coal-gas, the Boys Calorimeter is, perhaps, the best known and most widely used. Another type is the Simnance Recording Calorimeter for giving a graphical record of the calorific value. The Junker Calorimeter is also particularly applicable to fuel calorific measures.

For measuring the calorific value of a fuel, the gas type of calorimeter can be modified, and with suitable precautions very accurate measurements can be made.

The author has obtained satisfactory results with the Boys gas calorimeter previously mentioned, and, it may be added, that this was the method employed by Watson and Thomas in their extensive and accurate fuel tests of 1908-9.

The Boys Calorimeter.—This apparatus was devised by Professor C. V. Boys, F.R.S., and was described by him in a paper read before the Royal Society in 1905. It was afterwards adopted as standard by the Metropolitan Gas Companies, on the passing of the London Gas Act in 1906, and since employed by them.

Briefly, the principle upon which the apparatus works consists in burning the gas (or fuel vapour) at the burners with the correct amount of air for complete combustion, and in measuring the inlet and outlet water temperatures, and the quantity of water circulating through in a given time. The quantity of fuel consumed in the same time is also measured. Sufficient data are thus available for estimating the calorific value. Thus, if w lb. of fuel is burnt in t minutes, and if T_1 and T_2 be the inlet and outlet water temperatures centigrade, and W the weight of water in lb. flowing through per minute, then $T_2 - T_1$ = rise of temperature.

[1] For actual values and formulæ see p. 8.

Heat generated per min. $= W(T_2 - T_1)$ B.T.Us.

Weight of fuel burnt per min. $= \frac{w}{t}$ lb.

$\therefore$ Calorific value of fuel $= t \cdot \frac{W}{w}(T_2 - T_1)$ B.T.Us.

It is necessary to apply certain corrections for the condensed water of combustion, the specific heat of the water, and radiation.

The calorimeter is shown in vertical section in Fig. 43. It consists of three parts as follows: (1) The base A, carrying a pair of burners B and a regulating tap. The burners are made of steatite, and are carried by a head and tube also made of steatite. The upper surface of the base is covered with a bright metal plate, held in place by three screws and lifting blocks C. (2) A vessel D which is carried by the blocks C and is provided with a chimney E, and a condensed water outlet F. (3) The essential parts of the calorimeter, which rest on the rim of the vessel D; these are attached to the lid G. Beginning at the centre there is an outflow, and a brass box which acts as a temperature equalizing chamber for the outlet water.

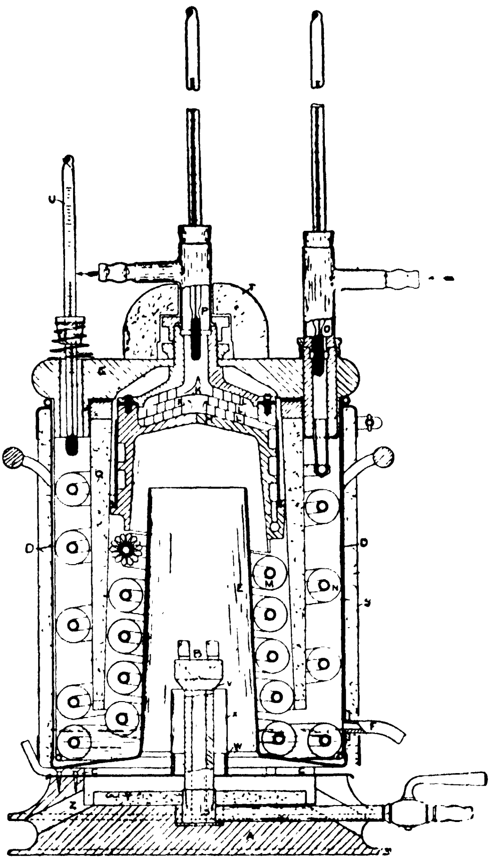


FIG. 43.—The Boys calorimeter.

Two dished plates of thin brass KK are held in place by three scrolls of thin brass LLL. The lower or pendant portion of the box is kept cool by water circulating through a channel which is cut in the wall of the bell. Connected to the water channel at the lowest part by a union are five or six turns of copper pipe of the Clarkson or earlier motor-car radiator type.

In this a helix of copper wire threaded with copper wire is wound round the tube, and the whole is sweated together by immersion in a bath of melted solder. A second coil of pipe of similar construction surrounding the first is fastened to it at the lower end by a union. This terminates at the upper end in a block to which the inlet water box and thermometer holder are secured by a union as shown at O. An outlet water box P and thermometer holder are similarly secured above the equalizing chamber. The lowest turns of the two coils MN are immersed in the water which, in the first instance, is put into the vessel D.

Between the outer and inner coils MN is placed a brattice Q, made of thin sheet brass containing cork dust to act as a heat insulator. The upper annular space in the brattice is closed by a wooden ring, and that end is immersed in melted rosin and beeswax cement to protect it from any moisture which might condense upon it. The brattice is carried by an internal flange which rests upon the lower edge of the casting H. A cylindrical wall of thin sheet brass, a very little smaller than the vessel D, is secured to the lid, so that when the instrument is lifted out of the vessel and placed upon the table, the coils are protected from injury. The narrow air space between this and the vessel D also serves to prevent interchange of heat between the calorimeter and the air of the room.

The two thermometers for reading the water temperatures and a third for reading the temperature of the effluent gases are all near together and at the same level.

A regular supply of water is maintained by connecting one of the right-hand pipes shown in Fig. 43 to a small tap over the sink. The overflow funnel is fastened to the wall about one metre above the sink, and the other outer pipe is connected to a tube in which there is a diaphragm with a hole about 2·3 mm. in diameter. This tube is connected to the inlet pipe of the calorimeter.

The thermometers for reading the temperature of the inlet and outlet water are divided on the Centigrade scale into tenths of a degree, and are provided with reading lenses and pointers that will slide upon them. The thermometers for the hygrometer and for reading the temperature of the air near the instrument and of the effluent gases are divided on the Fahrenheit scale into degrees.

The flow of air to the burners is determined by the degree to which the passage is restricted at the inlet, at the outlet, and at the base of the brattice. The blocks C, which determine the restriction at the inlet, are made of metal about 5 mm. thick, while the holes round the lid, which determine the restriction at the outlet, are five in number and are 16 mm. in diameter. The thermometer used for finding the temperature of the effluent gases is held by a cork supported on an open spiral of wire.

Application to Liquid Fuel Tests.—It is necessary when using

liquid fuels to obtain a perfectly regular supply of fuel, so that the rate of heat production is constant, and combustion always complete; further, no fractionation of the petrol must occur.

The arrangement shown in Fig. 44 works satisfactorily. It consists of a steam-jacketed vaporizing device, and a constant head device for maintaining a uniform flow of fuel. The fuel is delivered at a constant rate at the jet A, and, in falling, it strikes a series of

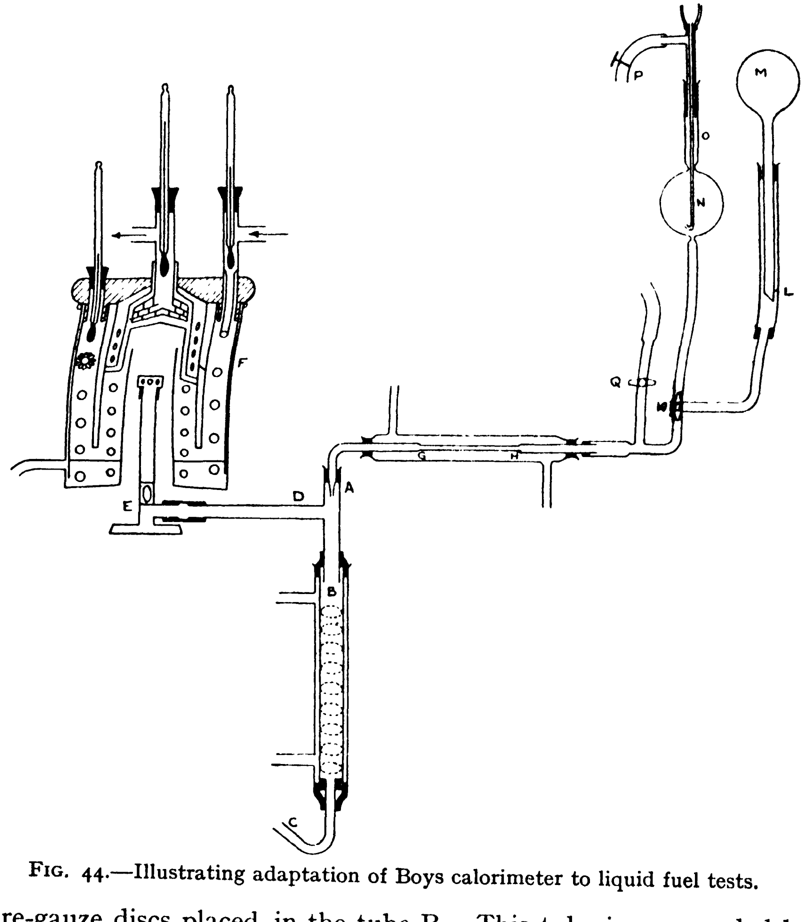


FIG. 44.—Illustrating adaptation of Boys calorimeter to liquid fuel tests.

wire-gauze discs placed in the tube B. This tube is surrounded by a steam jacket, and a gentle but steady stream of air flows in at C and passes B and D to the Bunsen type of burner E (which is substituted for the type B shown in Fig. 43). The air is so regulated that all the fuel is vaporized and carried through the tube D to the burner E. By turning the regulating cap at the base of the burner, the petrol vapour is caused to burn with a blue flame, almost free from the yellow cap. The hot gases of combustion circulate in the

calorimeter F, and yield up their heat to the water which is circulated at a constant rate through the " radiator " coils. The temperatures of inlet and outlet are noted, as before, and, from the known quantity of water passing, the calorific value can be calculated.

The method of obtaining a constant flow is by means of the constant " head " device shown on the upper right-hand side in Fig. 44. The fuel is supplied from the bulb M, the tube L serving to keep the head constant. When the test is to be started the tap K is turned so that fuel is delivered from the bulb N, the air which takes the place of the fuel that flows out of the bulb entering through the capillary tube O. In this way the head which causes the flow of fuel is always equal to the vertical distance of the lower end of this capillary above the jet A. The side tube and tap Q serve to remove any bubbles of air when filling the apparatus. The capillary tube GH through which the fuel flows is water-jacketed in order to maintain the temperature constant. The usual diameter of this capillary tube is 0·5 mm. (0·02 in.) and the length 15 cm. (6 in.). The bend in the tube C serves to trap any fuel if the supply is too rapid, and *the absence of any fuel in this trap may be taken as an indication that all of the fuel is being vaporized.*

The water condensed from the hot gases of combustion is collected from the condensed water outlet F (Fig. 43), and weighed. The heat given up by this quantity of water is deducted from the estimated (higher) calorific value in order to obtain the *lower calorific value.*

If w lb. of condensed water collect per lb. of fuel burnt in the calorimeter, then, since the latent heat of water is 966 B.T.Us. per lb., the total number of heat units to be deducted from the measured calorific value is $966w$ B.T.Us. A correction should also be made for the amount of heat supplied in the steam jacket, since this is added to the calorific value of the fuel. This added heat serves to supply the latent heat of the petrol, and slightly warms the mixture of vapour and air. By keeping the air current quite small, the error introduced here can be made practically negligible. Alternatively, this vaporizing tube may be heated electrically, and the outside lagged sufficiently to reduce radiation ; the electrical input, in watts, can then be ascertained, and an appropriate correction made.

Latent Heat of Fuel.—It is important to distinguish between the results of bomb and burner-type calorimeter results. In the former case the measured calorific value is lower by the latent heat of vaporization which is supplied from the heat of combustion, whereas in the latter case the heat required to vaporize the fuel is supplied separately. The measured calorific value is, therefore, greater in the latter case, and since, in the case of an engine, the exhaust products (or waste heat) supply this latent heat, this value

should be taken in all determinations. On the other hand, in direct injection engines, the latent heat must be supplied at the expense of the internal energy of the fuel, and the bomb-calorimeter calorific value then becomes applicable.

For petrol, the latent heat is less than 1 per cent. of the internal energy, but in the case of alcohol it represents about 4 per cent.

The latent heat values for different fuels are given in Table IX.

TABLE IX

Latent Heats of Vaporization of Fuels (Ricardo)

Fuel	Sp. Gr. at 15° C.	Lower Calorific Value. *B.T.Us./lb.*	Latent Heat at Constant Pressure (Atmosph.) *B.T.Us.* per lb.	Lower Calorific Value corrected for Latent Heat.* *B.T.Us./lb.*
Aromatic-free petrol . . .	0·718	19,080	133	19,200
"D" petrol	0·760	18,770	132	18,890
Heavy aromatics . . .	0·885	17,900	136	18,030
Paraffin	0·813	19,000	108	19,100
Hexane	0·685	19,250	156	19,390
Heptane	0·691	19,300	133	19,420
Benzene, 98 % . . .	0·884	17,300	172	17,460
Toluene, 99 %	0·870	17,520	151	17,660
Xylene, 91 %	0·862	17,800	145	17,930
Cyclohexane	0·786	18,800	156	18,940
Hexahydrotoluene . . .	0·780	18,760	138	18,890
Ethyl alcohol, 98·5 % . .	0·798	11,470	406	11,840
Ethyl alcohol, 95 vol. % . .	0·815	10,790	442	11,130
Methyl alcohol (wood naphtha) .	0·829	9,630	500 approx.	10,030
Methylated spirit . . .	0·821	10,200 †	450 approx.	10,580

* The lower calorific value plus the latent heat at *constant* volume.
† This value is probably correct for the ordinary purple variety of this sp. gr.

The Use of Bomb Calorimeters.—The method adopted in the case of the bomb calorimeter, which was designed primarily for solid fuels such as coal, is to burn or explode the fuel inside a calorimetric bomb, with oxygen, the bomb being placed in a double-walled vessel containing water. The rise in temperature of the known weight of water is measured and the calorific value (suitably corrected) is read off.

It is found, however, that when using volatile fuels, such as petrol and benzole, that complete combustion is difficult to secure, and, moreover, if the ordinary platinum crucible is used to hold the liquid fuel, it is very often shattered by the violent explosion. A better method, due to Berthelot, is to enclose the fuel in a relatively deep cup provided with a celluloid envelope, rising above the edge of the cup, and contracted at the top so as to form a kind of sack with

a relatively narrow mouth. In this way the vapour is confined sufficiently to cause it to burn at a moderate rate. It is necessary to apply a small correction for the heat of combustion of the celluloid.

The Griffin bomb calorimeter is shown illustrated in Figs. 45 and 46. In this apparatus the body of the bomb is machined from a solid forging of an acid-resisting alloy-steel. The inside is cylindrical in shape, and is provided with a round bottom ; the whole is highly polished. The top is of gun-metal and is gold plated inside to resist the corrosive influence of the combustion products. It is designed with an inlet and outlet, which can be opened or closed

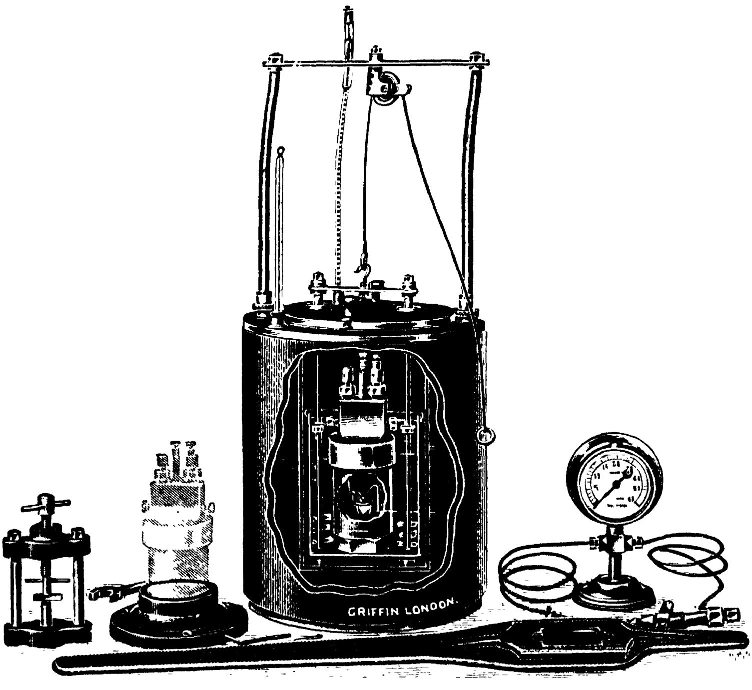


FIG. 45.—The Griffin bomb calorimeter.

by two specially designed valves. The feature of these valves is a self-centering conical plug which automatically finds its true seating without deformation. By providing two points of communication with the combustion chamber it is possible to withdraw the products of combustion for analysis in a series of weighed absorption tubes. The gas should enter through the valve communicating with the tube leading directly to the bottom of the bomb. The crucible containing the supply of fuel is supported on this tube. One end of the ignition wire which is placed in contact with the fuel is looped around the tube supporting the crucible, while the

other end is twisted round the insulated wire which leads to the centre terminal. The tube supporting the crucible and the insulated wire are of nichrome, although these may be of platinum. Beckmann thermometers in $\frac{1}{100}$° C. and also in $\frac{1}{10}$° C. are used.

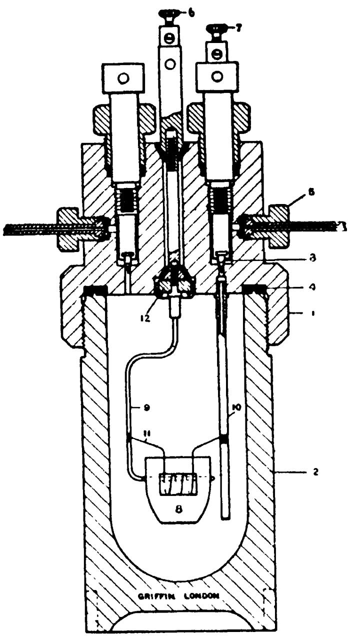


FIG. 46.—Explosion vessel of Griffin calorimeter.

The Junker calorimeter is another example which has been used in this country.

Computing Bomb Calorimeter Results.—It is necessary in computing the final results to apply a correction for the thermal capacity, or water equivalent, of the bomb, the thermometers, etc., since these absorb part of the heat of combustion, as well as the water.

The water equivalent of the calorimeter may be ascertained by burning a known weight of a substance whose calorific value is known very accurately. Naphthalene is a most suitable substance, having a calorific value of 9668 calories per gramme (1 gramme calorie = 0·00397 B.T.Us.).

In a certain test with this calorimeter 1·052 gramme of naphthalene was burnt, and gave a temperature rise of 3·130° C. The heat of combustion of the naphthalene was therefore 1·052 × 9668 = 10,071 calories.

The total water equivalent of the apparatus (and water)

$$= \frac{10171}{3{\cdot}130} = 3249 \text{ grammes.}$$

The weight of water in the calorimeter = 2500.

Therefore, the water equivalent of bomb, thermometer, etc. = 749 grammes.

The following is an example of the method of ascertaining the calorific value of petrol :—

Weight of petrol burnt = 0·732 gramme.

Temperature readings taken.

Time (mins.)	Temp. °C.	Time (mins.)	Temp. °C.	Time (mins.)	Temp. °C.	Time (mins.)	Temp. °C.
10·2	0·540	10·9	2·75	10·13	3·015	10·18	3·007
10·4	0·545	10·10	2·93	10·15	3·015	10·20	3·003
10·6	(fire)	10·11	2·98	10·16	3·015	10·22	2·997
10·7	1·24	10·12	3·01	10·17	3·011	10·24	2·994
10·8	2·22						

Temperature before firing	$= 0{\cdot}545°$ C.
Corrected [1] temperature after firing	$= 3{\cdot}043°$ C.
Rise in temperature	$= 2{\cdot}498°$ C.
Rise per gramme of petrol	$= \frac{2{\cdot}498}{{\cdot}732}\ 3{\cdot}415°$ C.
Water equivalent $= 2500 + 749$	$= 3249$ grammes.
Calorific value $= 3249 \times 3{\cdot}415$	$= 11{,}090$ calories per gm.
	$=$ 4,410 B.T.Us./gm.
	$=$ 20,000 B.T.Us./lb. app.[2]

The Junker Gas Calorimeter.—This calorimeter belongs to the continuous flow class and in its present developed form enables quick and accurate determinations of calorific values of gases and liquids to be made.

The amount of water flowing through the calorimeter in the time that a known volume of gas (or fuel) is burnt, and the resultant temperature change of the water, furnish the necessary data for calculating the calorific value of the gas.

The calorimeter body consists of a double-walled brass vessel into which water from a *constant level* pressure tank enters the inlet pipe F, flowing past the inlet thermometer R ; the water then flows through a micrometer valve G into the water chamber I. The heat produced by the burner, which is of the Bunsen type, with air and gas regulation, is conducted to the circulating water through the combustion chamber A and condenser tubes B. Thence the exhaust gases pass through an opening C, provided with a regulating valve D. The temperature of the gases is taken with the thermometer T.

The water passes through a series of baffle plates K at the top of the calorimeter where the outlet temperature is measured by means of the thermometer S, and through the outlet overflow weir L into measuring pails.

The various parts of the calorimeter in question are clearly shown in Fig. 47. In connection with the Bunsen burner a mirror is attached below it to permit observation of the flame during the operation.

Proportion of Air to Fuel.—The mixture strength can be determined by two independent methods. (1) By direct measurements of the quantities of air and fuel ; and (2) by exhaust gas analysis.

The methods of measurement of the quantity of air supplied form the subject of another chapter, so that we need only consider the methods of measuring the quantity of fuel consumed here.

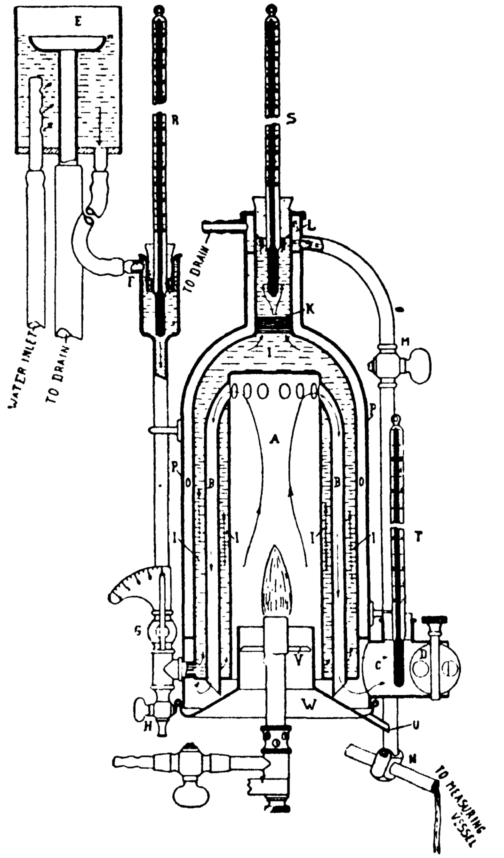
1. **Fuel Measurements.**—There are three principal methods employed for fuel measurements during petrol engine tests, namely :

[1] Corrected for radiation losses.

[2] 1 lb. $= 453{\cdot}6$ grammes.

(1) The constant volume-timing method; (2) the fuel weighing method; and (3) the flowmeter method.

The former method, which is the simplest and most direct, is usually employed for fuel measurements over relatively short test periods (of from 1 to 5 mins.), whereas the two latter methods are more convenient for continuous and duration tests; their use enables the average consumption to be ascertained.



A. Combustion chamber.
B. Condenser tubes.
C. Outlet for exhaust.
D. Damper for exhaust.
E. Inlet overflow weir.
F. Water inlet.
G. Inlet water valve.
H. Drain cock.
I. Water space.
K. Mixing device.
L. Outlet overflow weir.
M. Shut-off cock.
N. Change over device.
O. Air space.
P. Outer casing.
R. Inlet water thermometer.
S. Outlet water thermometer.
T. Exhaust thermometer.
U. Condensate drain.
V. Centering device for burner.
W. Detachable exhaust chamber.

FIG. 47.—A sectional view of the Junker calorimeter.

A simple method is that illustrated in Fig. 48, in which the main supply of fuel is contained in a tank T, and normally supplies the engine by way of the three-way cock C. A graduated glass vessel *ab* is also connected with the cock C, and it can be filled whilst the engine is running by turning the cock in an anti-clockwise direction by one-quarter of a turn. If the fuel level is too low in T for *ab* to be filled, pressure may be applied to the petrol as in the earlier automobile pressure feed supply, or the tank can be raised

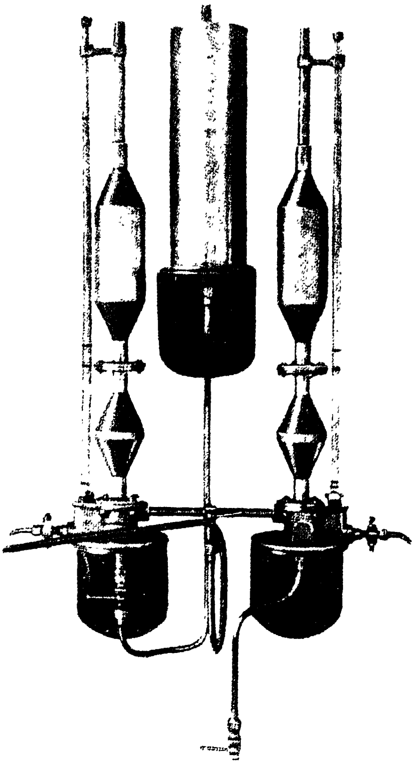


FIG. 40. The Ricardo fuel measurement apparatus.

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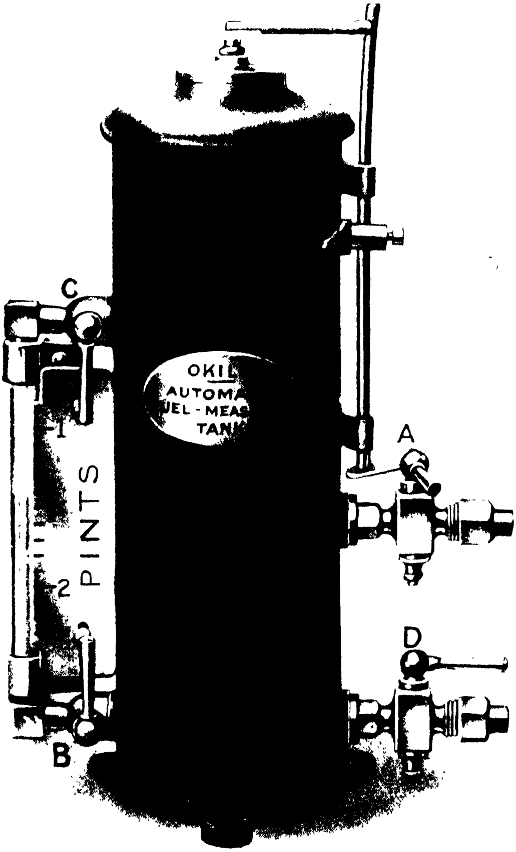

FIG. 50.—The Okill fuel-measuring device.

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temporarily. Normally the engine draws its fuel from T, but during the fuel consumption tests from *ab*. The initial level of the fuel is above the graduation *a*, in order to enable the engine to run for a time (before making a fuel test) from the supply in *ab*. The time taken for the fuel level to fall from *a* to *b* is measured with a stop-watch, and from its known volume and temperature the weight corresponding can be at once computed.

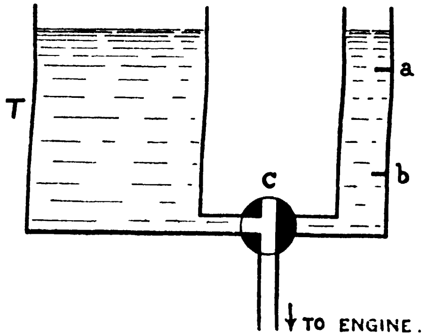

FIG. 48.

Thus, if the volume between *a* and *b* be denoted by V cubic inches, *t* is the time taken for this volume in minutes, and T° C. be the temperature of the fuel.

$$\text{Then fuel used per minute} = \frac{V}{t} \text{ cubic inch.}$$

$$\text{Weight of fuel per hour} = 2{\cdot}1701 \frac{V}{t} \,.\, d_t \text{ lb.,}$$

where d_t = fuel specific gravity at T° C.

An improvement upon the device described is that shown in Fig. 49, which represents the fuel measuring device used by Ricardo.

It consists of a pair of double conical vessels provided with a glass gauge on the side of each. The upper vessel holds exactly one pint between the marks on the gauge glass, and the lower vessel holds one-quarter of a pint. The object of the conical ends and narrow necks is to ensure that the fall of the fuel past the marks on the gauge marks is as rapid as possible, in order to minimize timing errors. Either of the two vessels can be used independently, and the engine can be supplied directly from the central tank. Each vessel can, therefore, be filled whilst the engine is running, so that no time is lost.

The constant volume fuel-measuring device can also be used in conjunction with the engine revolution counter, so that the total and average revolutions corresponding to the fuel timing period can be ascertained. This is accomplished by means of an electric-type counter which is thrown into operation when the fuel passes the first gauge mark, and out of operation when it passes the lower mark, by means of a simple switch device. Ricardo employed, for his well-known fuel tests,[1] a revolution counter driven off the camshaft and operated by a magnetic clutch. It was so arranged that the counter was thrown into operation as the liquid in the

[1] *Vide* footnote, p. 143.

gauge passed the first mark and out of operation when it passed the second mark, and a brake was applied to the counter's spindle to prevent its over-running when thrown out of operation.

In this manner it was possible to obtain the actual number of revolutions corresponding to the known fuel volume consumed (i.e. one pint or one-quarter pint).

Capacity of Fuel Vessel.—The capacity of the fuel-measuring vessel should be chosen to suit the engine under test, and in this respect it is advisable to arrange for a fuel-measuring time of not less than 1 minute.

The average fuel consumption of a petrol engine of good design, working on the best combustion mixture, may be taken at 0·55 pint per B.H.P. hour, so that the approximate capacity of the measuring vessel can at once be ascertained.

Example.—Required the approximate capacity of a fuel-measuring vessel for a 30 B.H.P. engine.

The fuel consumption per hour $= 30 \times \cdot 55 = 16{\cdot}5$ pints approximately.

The fuel consumption per minute $= 0{\cdot}275$ pint.

Assuming a timing period of 2 minutes the required capacity is therefore $2 \times 0{\cdot}275 = 0{\cdot}55$ pint.

It is interesting to note that for the timing period of 2 minutes, the same value is used as for the fuel consumption per hour of the 30 h.p. engine.

The Okill Fuel Measuring Tank.—Fig. 50 illustrates the Okill automatic fuel consumption device. It consists of a reservoir of suitable capacity for a minimum test run of 4 minutes, and is fitted with inlet and outlet cocks A and D respectively, and gauge glass cocks B and C. There is a float operated valve at the top which allows the air to escape when filling and to enter when the engine is running on fuel from the measuring tank ; this valve also controls the amount of fuel admitted to the tank, automatically and accurately. The fuel consumption of an engine is measured by starting a stop-watch (or noting the time) when cock A is closed, and by stopping the watch (or noting the time again) when cock B is closed at the end of the test. The amount of fuel consumed in this time interval is read off on the oil-level of the gauge. The cocks A and B are then opened and the measuring tank allowed to refill. The measuring tank is always ready for use, and does not interfere with the ordinary running of the engine ; all four cocks are normally open, during normal running, when not making a test.

2. **Fuel Weighing Method.**—In this method the supply fuel tank is carried on the bed of a weighing machine, the beam of which is usually graduated in pounds and tenths, and reads by estimation to one-hundredths. Readings are taken of the time for a definite weight of fuel to be consumed.

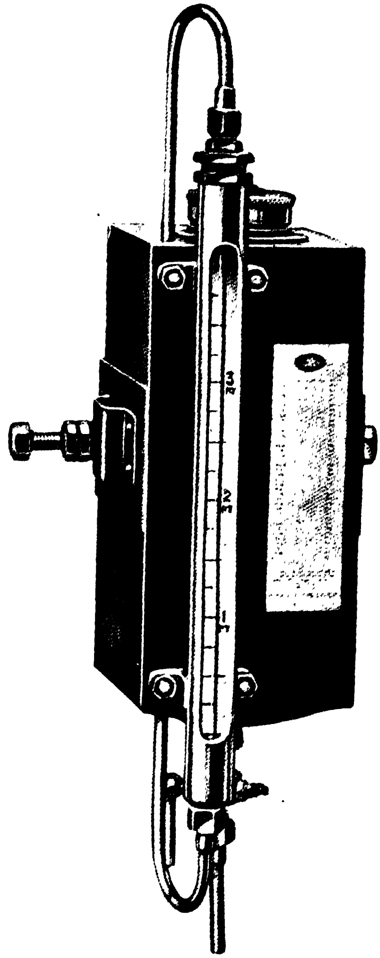

FIG. 51.—A convenient fuel-measuring tank for motor vehicle road tests. This has a half-gallon capacity and is graduated in pints.

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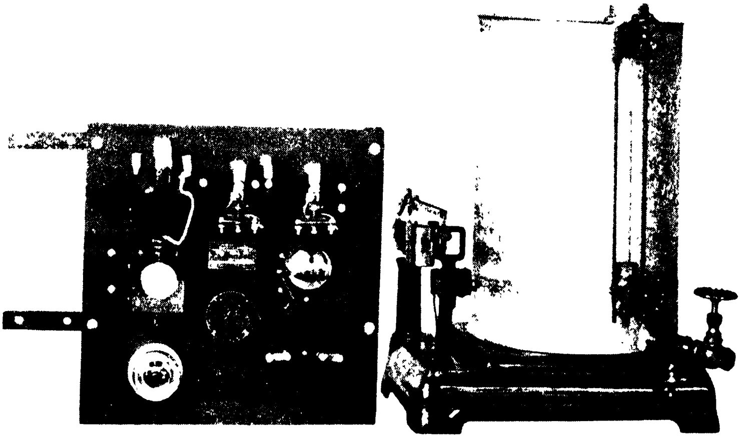

FIG. 52.—The Sprague-type fuel-measuring device and electric control panel (G.E.C. America).

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The tank and its contents are first weighed, and the poise weight set to the nearest whole pound reading below this weight. As soon as the fuel used causes the scale lever to drop, a stop-watch is started, and the weight slid back to a reading 1 lb. lower. When the scale lever rises the stop-watch is stopped and the time interval corresponds to that for 1 lb. of the fuel consumed by the engine.

An elaboration of this method, first employed by the Automobile Club of America, consists of automatic means for stopping the watch when the exact weight of fuel (decided upon beforehand) has been consumed. In this case the scales are wired to a battery in such a way that a point on the scale beam dips into a mercury cup when the scale beam falls, thus closing a circuit which rings a bell and stops the electrically-operated watch. To illustrate the method imagine that just before starting a test the tank and contents weigh a little over 65 lb. With weights and rider set at exactly 65 lb. the beam remains up. As soon as sufficient fuel is consumed, the weight decreases, and when 65 lb. is reached the beam drops. The stop-watch is started by hand at this instant. The rider weight is now slid back to 64 lb. and the beam rises. The observer closes the switches in the bell and watch circuits and can then leave the apparatus to attend to other work. As soon as the weight falls to 64 lb. the scale beam drops, closes the electrical circuits, and thus stops the watch and sets the bell ringing to attract the observer's attention. The observer then sets the scale to 63 lb., enters the elapsed time on the log, and repeats the procedure. The use of a split-seconds hand stop-watch enables both continuous and interval period times to be taken, and is a great advantage in this case.

The American Altitude Laboratory of the Bureau of Standards employ the fuel-weighing device, as an alternative to the constant volume one. When using the former method, two tanks are generally used, so that a test may be run continuously, one tank being filled while the other is emptying, or two fuels may be compared as follows: One tank is filled with the fuel to be tested, and the second with a standard comparison fuel. The engine is first run on the standard fuel and is then changed over to the fuel test, after which a third run is made on the standard fuel.

The Sprague fuel-metering device illustrated in Fig. 52 is also of the automatic type, and is used with an electrically-operated speed counter and stop-watch. It consists of a six-gallon fuel tank provided with a sight-glass and fittings. This tank is carried on a 50-lb. platform scale with electric contacts of the "make" type on the beam. A panel, with a stop-watch and relays for starting and stopping the watch, and an electric speed-counter, is provided as part of the apparatus, but is mounted separately to the metering device, in any convenient place in the test laboratory. A 6-volt

dry battery is mounted on the back of the panel, but the main relays are operated on 115 or 230 volts D.C.

An account of the operation of this device will explain the principle. The engine is run at the desired speed and load for the fuel consumption test, and the main switch in the relay circuit is closed. The stop-watch is set to zero by means of a lever provided.

Next, the rider weight on the scale beam is set to a value less than the total weight of the tank and fuel, so that the beam is against the upper stop ; the double throw switch (on the left of the panel) is closed so as to connect up the electric circuits for operating the watch, speed counter, and a warning bell. It is advisable to record the speed-counter reading at this stage, and also, in the case of brake tests, the dynamometer scale reading. As soon as sufficient fuel has been consumed the weight on the weighing platform falls to the value to which the rider weight was set (say 35 lb. total), and the beam falls. In falling it starts the watch and also the electric speed counter, and rings a warning bell. The rider weight is now set back to a lower value (say 30 lb. total). When the equivalent weight of fuel (5 lb. in this case) has been consumed by the engine, the beam again falls, and in doing so stops both the watch and the counter, and also rings the alarm bell for the second time. The stop-watch and counter readings are recorded down, together with that of the dynamometer scale, and sufficient data are then available to enable the fuel consumption per B.H.P. hour and the average speed to be ascertained.

From the time readings, the time to consume a given weight (say 5 lb.) of the fuel can be ascertained.

The difference between the initial and final speed-counter readings divided by the stop-watch time interval gives the average speed of the engine, whilst the dynamometer torque reading and this speed value enable the B.H.P. to be computed.

In connection with the weighing method for fuel consumption, this is particularly suitable for duration, or long period consumption tests, but it is not so accurate nor convenient as the constant volume method for frequent individual tests of short duration.

Fuel Flowmeters.—The methods hitherto considered give the average fuel consumptions over a pre-arranged period of time, or for a given total consumption. It is frequently necessary to be able to tell exactly what the fuel consumption is at any moment, so that for this purpose the preceding devices are not suitable, and a continuously reading instrument or apparatus is required. The analogy of the average speed, obtained by dividing the total revolutions of the counter by the time interval, applies to the methods hitherto described, whereas the instantaneous reading of the tachometer applies to that of the flowmeter, or instantaneous fuel consumption device.

There are two principal kinds of fuel flowmeter used for engine tests, namely, the *venturi* and the *variable orifice* ones.

The principle of the former flowmeter as adapted for use by the U.S. Air Service [1] is illustrated in Fig. 53. The parts consist of a venturi tube, a differential pressure gauge, and connecting tubing. The venturi tube is inserted in the fuel line leading to the carburettor and, as is shown in the figure, is connected with the indicator by means of two copper tubes of small bore.

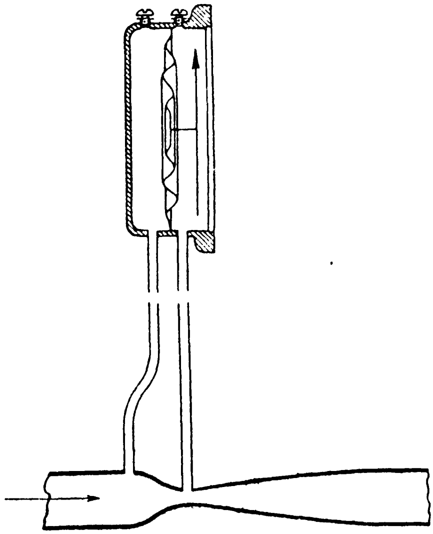


FIG. 53.—Venturi-type of flowmeter.

One tube connects the throat section of the venturi tube with one side of the diaphragm, or, more exactly, to the interior of a diaphragm capsule, and the other connects the entrance section of the venturi to the case of the indicator. When fuel flows the fuel pressure is less at the throat section than at the entrance, which difference is indicated by the gauge.

It can be shown, analytically, that the relationship between the pressure difference in the two (vertical) tubes P, the average volume rate of flow Q, and the density D of the fuel is as follows :—

$$P = \left(\frac{Q}{M}\right)^2 \frac{D}{2g}$$

where M is a constant depending upon the dimensions of the two venturi cross-sections and g is the acceleration due to gravity.

A high value of M is obtained by making the cross-sectional area ratios as large as possible. The value of the fuel density, it will be observed, affects the flowmeter readings so that an account should always be made of the actual value of the fuel density, or its temperature—upon which the density depends. An objection to this type of flowmeter for aircraft purposes is that if the instrument is placed on the pilot's panel there is an added risk of fire in the case of accident.

A difficulty often experienced is that of keeping the lines to the indicator and pressure capsules entirely filled with fuel ; suitable

[1] "Aircraft Power Plant Instruments," Harcourt Sontag and W. G. Brombacher, N.A.C.A. Report, No. 466.

priming, "bleeding" or venting arrangements must therefore be provided.

The principle of the Amal flowmeter is shown diagrammatically in Fig. 54. A and B are vertical tubes connected at their lower ends by a venturi passage V. A constant level of petrol is maintained in A by means of the float feed mechanism shown at C; the level in A is thus kept constant, independent of the rate of flow through the venturi V. An outlet at D is controlled by a stop-cock. It will be evident that when the latter is closed fuel in B will reach the same level as in A, but if the stop-cock is open, the level in B will fall to some point at which it will then remain stationary. The difference in the levels of A and B depends upon the rate of flow of the fuel, so that we have here a means of measurement. The range of difference in the height of the fuel in A and B is in its turn controlled by the discharge, under constant head, through the venturi orifice, which in this instrument takes the form of an accurately calibrated jet.

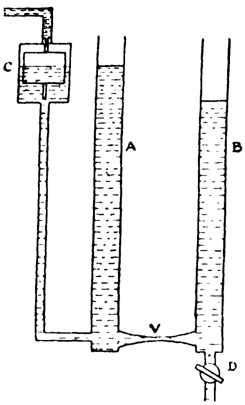


FIG. 54.

The formula for the scale graduations is

$$Q = C\sqrt{2gh}$$

or

$$h = \frac{Q^2}{C^2 \cdot 2g}$$

where Q = actual cubical discharge from the delivery side D,
h = difference in height between A and B,
C = a constant, or discharge coefficient for the venturi orifice which varies with the area of the orifice and its design. Its value is experimentally determined in each case.

The constant C can be readily found by measuring the total quantity Q of fuel discharged through D in a given time, and at a known temperature.

The more recent Amal flowmeters,[1] based upon the principle described above, are made in three chief types, viz. the Mark VIII, Mark IX, and the Mark XI; the corresponding maximum capacities are 54, 480, and 960 pints per hour, respectively.

Each flowmeter is provided with two ranges, and two identical, or different, scales can be used with each instrument.

The flowmeters in question can be supplied graduated in English, Metric, or American units.

[1] Messrs. Amal Ltd., Birmingham.

The result of a considerable number of tests made with these flowmeters has shown that the accuracy of these instruments is within 1 per cent.

The flowmeters have been adopted for official tests of aircraft engines (bench tests) and are now widely used in the test shops of automobile manufacturers.

In regard to motor car and cycle engine tests the Mark VIII instrument is the most suitable.

Fitting and Using the Flowmeter.—When fitting up flowmeters of this type it is important to note that they should be fixed rigidly in a vertical position, the head of the flow to the instrument being not less than 9 inches or more than 18 inches as measured from the top of the float-chamber. The scale is calibrated for a standard grade petrol at 15° C. To use the flowmeter have the cocks on the instrument closed, then turn on the supply from the main, and wait until the instrument is filled ; then open the air release valves and afterwards the petrol cocks. Leave the air release valves open until petrol flows from them, then close.

Leave cock fully open, and continue tests in the usual way. The instrument will then record the amount (i.e. rate of flow) of petrol passing through it in the units shown on the calibrated scale.

If the petrol or other fuel in use is not well filtered it is advisable to remove the two venturi jets occasionally, and flush out the venturi body to clear it of sediment, etc.

Whilst the jets are out of the machine a piece of soft rag moistened in spirit should be used to remove any silting of the orifice, which, as it collects, will tend to cause the instrument to indicate a larger flow than is actually taking place.

No metal instrument of any description should be put into the orifice.

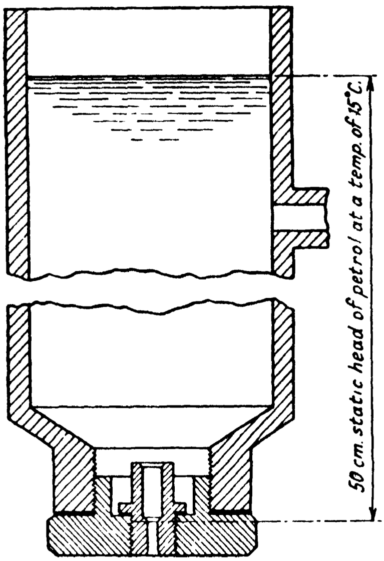
As the instrument is constructed to damp out the major portions of the violent fluctuations which sometimes occur in various float chambers, no reading should be taken until the instrument has been recording at least one minute under the desired conditions. Never attempt to run the engine when the instrument indicates more than the maximum on scale.

Jet Calibrating Apparatus.—A carburettor jet calibrating machine is also made by the same Company. This apparatus enables the calibration of a jet to be made in a few seconds ; it reads the discharge that would take place with a standard jet at the standard head, temperature and density.

The standard is a jet (Fig. 55) having a smooth regular restriction of not less than 3 diameters long or more than 5, under a static head of 50 centimetres measured from the commencement of the orifice, the temperature being 60° F., and petrol density ·710. The

discharge in c.c. per minute is taken as the standard of comparison with other jets.

Fig. 57 shows the Mark X jet calibrating apparatus. The complete instrument is mounted on a backboard, which should be firmly fixed in a vertical position. The petrol passes from tank A, through filter B and tap C, to float chamber D ; thence through tube to the body of the instrument, and rises in glass tubes F and F1. The jet to be calibrated is placed in adapter K or K1 and tap H or H1 is turned on by rotating same. The air above the jet is released by means of screws L or L1, and the level of the petrol F or F1 is read off on scales G. The petrol flowing from the jet is collected



Standard jet length of restriction = 4 diameters
Head is measured from commencement of restriction

FIG. 55.—Standard jet for Amal flowmeter.

through tray M in tank N, and is returned to top tank A, through pipe P, after closing tap R and air release S, by the operation of the pump O.

If a jet be now fixed in either of the adapters K and K1, and the corresponding cock H and H1 opened, the petrol in the gauge glass above it will fall to a level varying in accordance with the size of the jet, and it is this level which, when read off on the scales G, gives the calibration of the jet ; the level in the gauge glass not in use will remain practically at zero. To correct for barometric temperature and density variations, a standard or master jet is supplied with each instrument, to which each scale should be set. This jet should be inserted and the petrol allowed to flow for at

FIG. 56. The Amal flowmeter. Mark VIII type.

[See page 103.

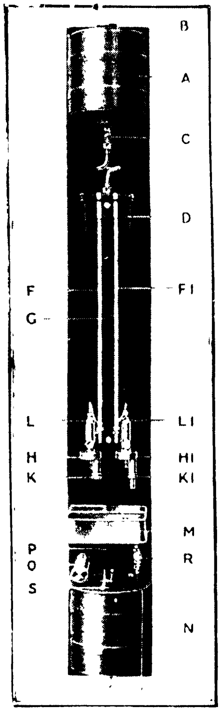


FIG. 57. Carburettor jet calibrating apparatus.

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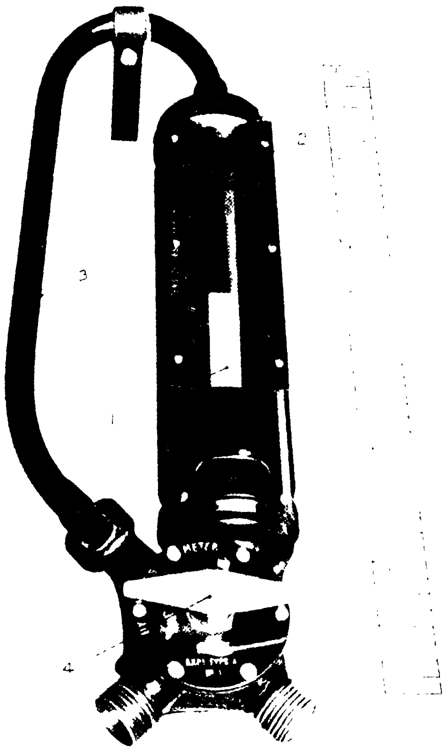

FIG. 59. Experimental mass-flow measuring flowmeter.

[To face page 105

least one minute when the mark on the scale corresponding to the master jet size should register level with the petrol in the glass gauge tube.

It should be clearly understood that the master jet is used to set the particular scale which is being used at the time, that is to say: if there is a scale of 20 to 100 on one side, and 100 to 500 on the other, a master jet of say 100 c.c. should not be set on the 100 line on the smaller scale, if the larger scale is being used, and vice versa.

By this means all variations due to temperature, density, etc., are automatically corrected. As a shop is likely to vary in temperature throughout a normal day, for accurate calibration these scales should be periodically checked over, say every two or three hours, in this way.

FIG. 58. — The Elliott flowmeter.

The jet calibrating apparatus described differs from the flowmeter previously mentioned as the head under which the jet would be tested in the latter case would be variable for each size of jet.

The apparatus in question is made in a size suitable for motor cycle and car engines. In this case the L.H. scale is graduated from 20 to 100 c.c., and the R.H. one from 100 to 500 c.c. There are also three ranges for aircraft engines from 100 to 5000 c.c.

The Amal flowmeters can be supplied with a fuel tank mounted on a backboard with a convenient arrangement of two-way cocks to enable the *rate of fuel consumption of a motor vehicle* to be measured under actual road conditions. It has been designed for use by car and commercial vehicle manufacturers.

The Elliott Flowmeter.—This instrument is designed primarily for aircraft, but it is equally applicable to ground tests of other internal combustion engines.

It gives an accurate measurement of the weight, in pounds, of fuel consumed per hour, although as its name indicates it actually reads the *rate* of *fuel flow*, or consumption.

Fig. 58 shows the appearance of the instrument, which consists of a conical tube through which the fuel flows, entering at the lower or smaller end, and leaving at the top or larger end. A sink is so arranged that its density is somewhat greater than the mean density of the fuel, and moves freely up and down a central guide. The fuel flows through the annular space between the conical tube and disc on the top of the sink. By reason of the fact that this disc is bevelled to a sharp edge, the flow past is turbulent, and consequently the readings are, for practical purposes, independent of the viscosity of the fuel due to temperature variation or by the effect

of the release of gases at high altitudes. Moreover, different grades of fuel may be used without the necessity for re-calibration. A pointer indicates the rate of flow against a uniformly graduated scale. An air release cock is fitted at the top of the instrument.

It is claimed that this flowmeter will read correctly to within 0·25 per cent. over a fuel density range of $\pm$ 10 per cent. from a mean value and to within 1 per cent. of the scale reading.

The flowmeter is inserted in the gravity feed pipe of the petrol system, but when used in a pressure pipe line the loss of head across the flowmeter will not exceed 3 inches of water.

In general, it is only necessary to use the air release cock during the initial priming of the pipe system and in cases where, following a shortage of fuel, it is necessary to clear away any possible air lock at the pipe bend.

The flowmeter is made in three standard sizes, viz. 50/150, 100/300 and 150/400 lb. per hour.

In order to express the flow in gallons per hour the readings in pounds per hour should be divided by the number of pounds per gallon of fuel, as found with an hydrometer.

Mass-Flow Measuring Flowmeters.—It is important for aircraft engine test purposes to have a flowmeter which will measure the actual mass-flow of fuel taken by the engine under flight conditions.

To give accurate readings such a flowmeter should be uninfluenced by variations of density changes due to temperature variations over the range of temperature likely to be met with in flight.

Experimental work carried out at the R.A.E., Farnborough,[1] has led to the design of a mass flowmeter for which an accuracy of $\pm$ 1·0 per cent. over a temperature range of $\pm$ 18° C. to $-$ 10° C. is claimed.

A later instrument (shown in Fig. 60), when tested with aviation mixture and also with kerosene, showed agreement of readings to within 1·0 per cent. This is equivalent to a test with aviation mixture over a temperature range of $+$ 25° C. to $-$ 30° C.

Flight tests with this type of flowmeter showed an overall accuracy within $\pm$ 2·0 per cent. at all altitudes up to 16,000 ft. Fig. 59 shows one form of mass flowmeter that gave the former results mentioned. In this instrument (1) is a sink which fits in a glass tube, with a clearance sufficient to permit grit and other foreign matter in the fuel to pass without restricting the free movement of the sink. The sink is provided with a central sharp-edged orifice through which the main flow passes. A tapered rod (2) passing through the orifice varies the area of the orifice with the height of

[1] "Fuel Flowmeters Designed to Measure Mass-Flow," P. S. Kerr, Aeron. Research Comm. R. and M. No. 1245 (E. 31), Jan. 1929.

the sink in the tube. The section of the rod is such that the scale of flows is uniform.

The fuel enters by the union on the right, passes up the glass tube, and returns *via* the pipe (3) to the union on the left. The cock (4) is provided with a by-pass position to isolate the instrument in the event of breakage of the glass tube.

The scale in "Tens of lbs. per hour" is on the left, and in action the position of the top of the sink in relation to the scale is read. The scale on the right is in inches, and is used for calibration purposes.

From theoretical considerations it can be shown that the density of the sink should be equal to twice the mean density of the fuel.

In the later model shown in Fig. 60 the small errors due to change of viscosity with temperature and sticking at low temperature were overcome. In this type the variable orifice is the annular space between the outer tube, which is conical, and a concentric sharp-edged disc fixed to the sink. This construction eliminates the viscous flow in the clearance between the sink and the tube, which was the chief source of error in the first design.

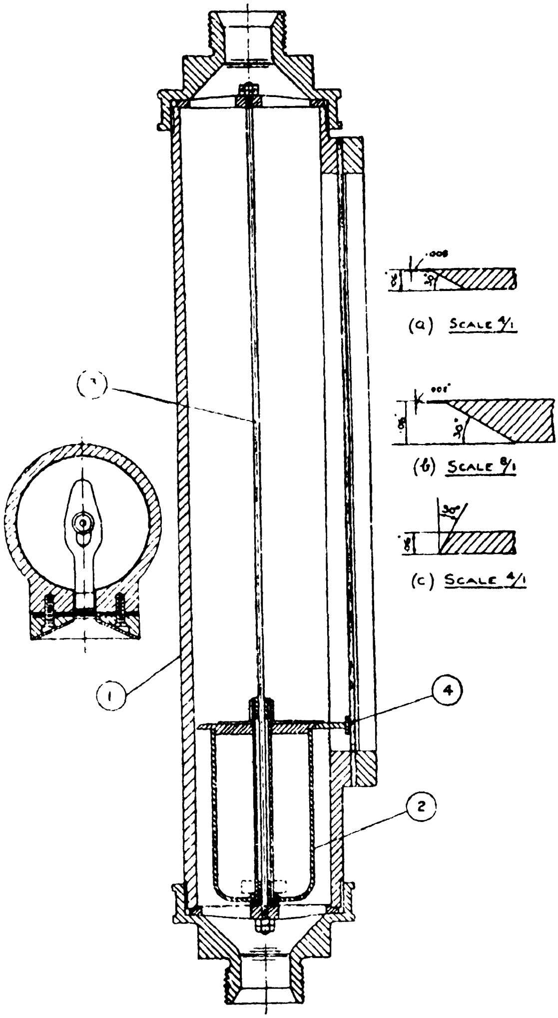

FIG. 60.—Improved type mass-flow measuring flowmeter.

In the section of the modified form of the instrument shown in Fig. 60, 1 is a brass tube tapered outwards from bottom to top in

order to provide the variable area of orifice. The sink (2) is kept in a central position relative to the tube but a guide rod (3). A narrow slot covered with cellon, and a tongue (4) projecting from the sink, enable the height of the sink in the tube to be observed.

Experiments with three forms of edge on the disc at the top of the sink were made. The forms are shown enlarged at (*a*), (*b*), and (*c*), Fig. 60. Form (*c*) was found to be most suitable for the reduction of the error due to change of viscosity with temperature.

Other Types of Flowmeter.—Flowmeters are made both in the vertical tube or vessel design, and also in a circular dial form ; the latter is more convenient for portability, but, as a rule, is not quite as accurate as the type previously described.

The Hamill flowmeter was of the latter type, and it has been applied to the testing of fuel consumptions on motor-cars. By means of an additional scale setting for the road speed (i.e. speedometer reading), the functions of this flowmeter and the speedometer may be combined and the consumption of fuel per road mile (expressed actually in miles per gallon) read off.

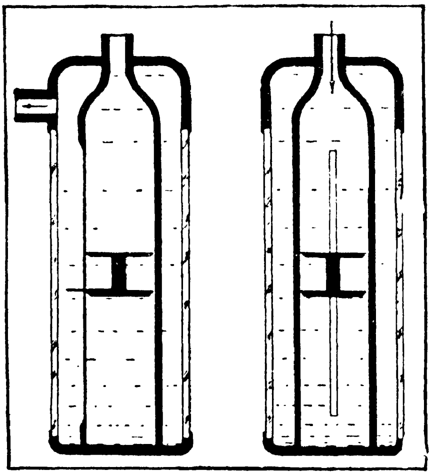

FIG. 61.—Automobile engine flowmeter.

Recording Flowmeter.—Any of the flowmeters described can be made self-recording by the simple expedient of employing a photographic recorder of the bromide paper and chronometer drum type. A light float and styles could be arranged to record the liquid level upon a paper drum, or by a simple optical or photographic means to record its positions by means of a beam of light impinging upon a roll of bromide paper moved regular by a clockwork mechanism.[1]

The American Bureau of Standards have employed a photographic method for obtaining flowmeter records when testing an automobile on the road. The type of flowmeter employed is shown diagrammatically in Fig. 61. It consisted of a vertical tube slotted parallel to its axis, and provided with a light piston free to move vertically in the tube. Fuel entered the tube at the top, flowing freely out of the slot below the piston into the ring-shaped space through which the position of the piston can be seen, and thence out at the top, as shown. A pointer affixed to the piston moved

[1] *Vide* also "Radium Recording Devices," *Journ. of Scientific Instruments*, April, 1924.

over a vertical scale, thus indicating the effective length of the slot. Knowing this length and the pressure, the rate of flow is obtained.

Fig. 62 illustrates the mechanism of the flowmeter camera. A strip of bromide paper was made to travel from one spool to another past the flowmeter, and a beam of light from the lamp shown just above the lower spool was reflected by a small mirror (on the extreme right) and thence past the flowmeter needle to the bromide paper. The pointer of the flowmeter throws a shadow on the paper, the area around which is exposed to the light, and thus becomes blackened, leaving the pointer shadow as a white region which the motion of the paper spreads out into a curve or line. Time intervals were controlled from a contact mounted on the tachometer of the car, and these were impressed on the fuel flow film by the alternating increase and decrease in the intensity of the illumination on the bromide paper.

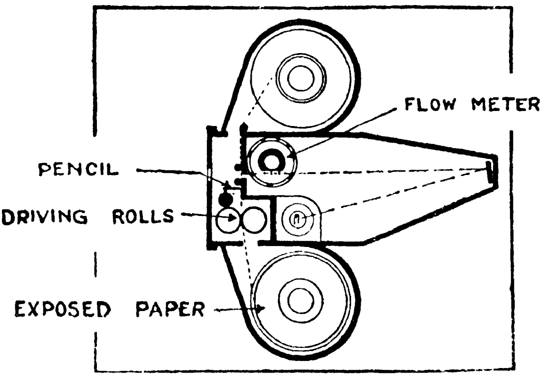

FIG. 62.—Automatic flowmeter recorder.

Road Fuel Consumption Devices.—For road tests of automobiles, special instruments have now been designed which enable the fuel consumption to be read off in miles per gallon. Of these the Hamill, Gallometer, Zenith and Solex are examples.

The Gallometer device, illustrated in Fig. 63, consists of a glass tube enclosed in a metal casing, provided with a slot through which the fuel level can be seen. The slot is graduated, as shown, with a miles-per-gallon scale. The device is used as follows: It is first filled with fuel by turning the three-way tap, below, to the filling position, until a slight overflow indicates that it is full. When the car passes a milestone, or when the speedometer mileage figure reads a round number of miles, the tap is turned over so as to cut off the main fuel supply, and to feed the carburettor from the glass reservoir direct. At the end of the next mile travelled by the car the tap is turned off and the fuel level in the glass vessel reads the m.p.g. directly.

In this case the volume-graduations are inversely proportional to the m.p.g. consumptions.

Zenith Fuel Consumption Device.—The Zenith Mileage Tester shown in Fig. 64 enables fuel consumption tests to be carried out on the road under actual running conditions. It consists of an accurately calibrated and graduated glass vessel of one-tenth gallon capacity, an electric fuel pump and a three-way cock mounted as a single unit and arranged for fixing in a convenient position for the driver, i.e. on the off-side window of the vehicle; vacuum cups and clips are provided for this purpose.

The two rubber pipes from the apparatus are connected to the

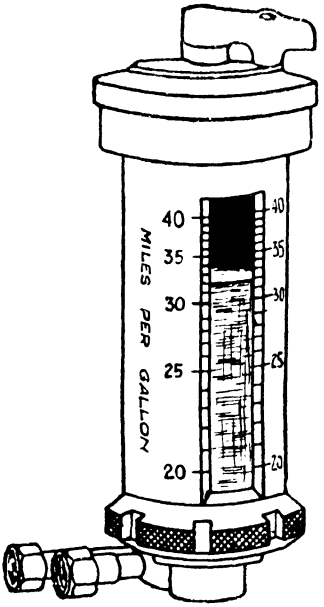

FIG. 63.—The Gallometer automobile fuel consumption device.

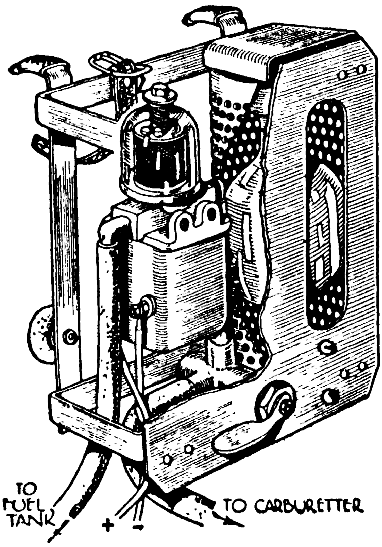

FIG. 64.—Zenith fuel consumption apparatus.

carburettor and fuel tank connections, respectively. The petrol pipe from the fuel tank or pump is disconnected at the carburettor. The pipe leading from the glass bulb to the tester is then connected to the carburettor whilst the other one from the pump is connected to the pipe from the tank or pump. Cars having a mechanical or electric fuel pump should have the pipe disconnected at the carburettor so that the tester is fitted between the pump and carburettor. The electric fuel pump of the tester is connected to the vehicle's electrical system in a similar manner to ordinary electric pumps.

In use the handle of the three-way cock is placed vertically when the fuel is pumped direct from the main tank to carburettor for normal driving requirements. When turned to the right fuel

is not only pumped direct to the carburettor but also to the glass bulb. When the fuel fills the latter to the proper level the cock handle is turned to the extreme left; this shuts off the main fuel supply to the carburettor and connects the fuel in the glass bulb to the latter. It is then necessary only to observe the speedometer readings when the fuel passes each of the two marks on the glass bulb. The speedometer mileage difference multiplied by ten gives the fuel consumption in miles per gallon for the car speed at which the test is made.

The Solex Fuel Consumption Tester.—This apparatus, which also acts as an emergency petrol tank consists of a small reservoir mounted on the back of the dashboard under the bonnet and is connected to the carburettor float chamber inlet pipe by means of a two-way cock. The tank is made in two sizes, namely, 1 and 2 pints, and has its own petrol cock. For gravity feed this auxiliary tank is fitted above the level of the carburettor; for pump feed a special cock union is provided so that the tank can be supplied by the pump and then disconnected. During a fuel consumption test the fuel supply from the pump to carburettor is shut off. The speedometer readings corresponding to the commencement and end of the run, at constant car speed, from the auxiliary tank give a means of estimating the mileage per gallon.

Combined Fuel Consumption Devices.—Hitherto we have described the constant volume, the fuel weighing and flowmeters as being in separate classes, but it should now be mentioned that either of the former two can be combined with the flowmeter, and that for general test purposes this is an advantage. The total fuel consumption over a given time period, as given by either of the former two methods, enables the average fuel consumption to be ascertained. By arranging a flowmeter between the fuel consumption tank (or vessel) and the engine, the instantaneous consumption can be checked at any time, and, if desired, the flow can be varied by aid of the flowmeter so as to give any desired fuel flow or mixture strength.

This method was adopted in some aircraft engine tests at Farnborough. Fig. 65 illustrates a rough layout of the apparatus in question. It will be seen that there was a main tank (of 105 gallons capacity) provided with a cock A for shutting off the main supply, and a cock B to control the supply to the consumption tank and also the straight-through supply to the engine. A large gauge-glass, graduated in gallons, was connected to the main petrol tank as a level indicator.

For convenience of ascertaining the rate of consumption at any time, two means were provided. The first consisted of a flowmeter, namely, a Venturi tube with a very sensitive pressure gauge to indicate the differences of fuel head pressure in the tube. The

second consisted of a diaphragm communicating with a dial, which was graduated to read the rate of consumption directly. A constant head to the Venturi tube was ensured by interposing a small tank with a ball-cock between it and the main tank. The vertical gauge in the centre was for indicating the velocity of the air blast on the engine. The average fuel consumption was ascertained, in the usual way, by switching over a three-way tap to feed the fuel from the consumption tank to the engine.

Testing Fuel Pumps.—It is occasionally necessary to test fuel feed pumps for satisfactory operation not only in connection

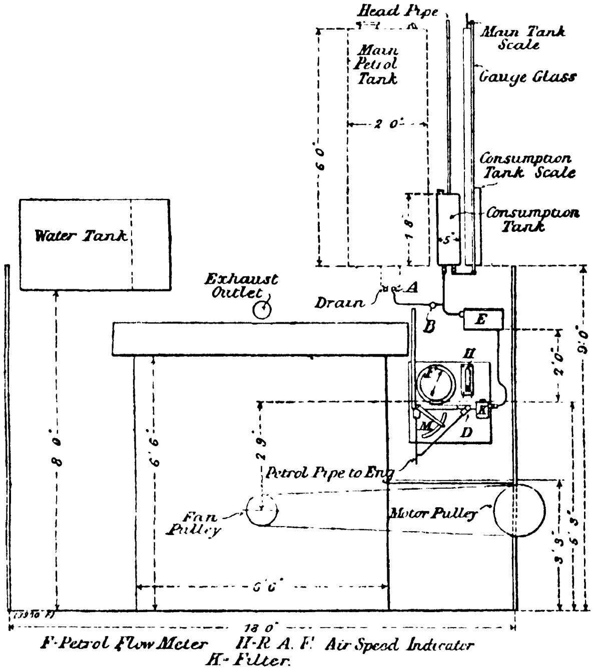

FIG. 65.

with engine tests but also for servicing purposes. The A.C. fuel pump testing apparatus, shown in Fig. 66, enables the operation and performance of a fuel pump to be investigated in a convenient and expeditious manner. The pump is mounted on the test stand and attached firmly by means of the wing nuts and bolts provided. The petrol hose pipe is connected to the inlet and outlet sides of the pump and, in the case of mechanically operated diaphragm type pumps, the actuating rocker is worked by hand. If the pump will commence to deliver fuel with twenty strokes of the rocker arm, or less, the shut-off valve on the outlet side is closed and the rocker arm is then actuated until a fuel pressure of $1\frac{1}{2}$ to $2\frac{1}{2}$ lb. per square inch is indicated on the pressure gauge.

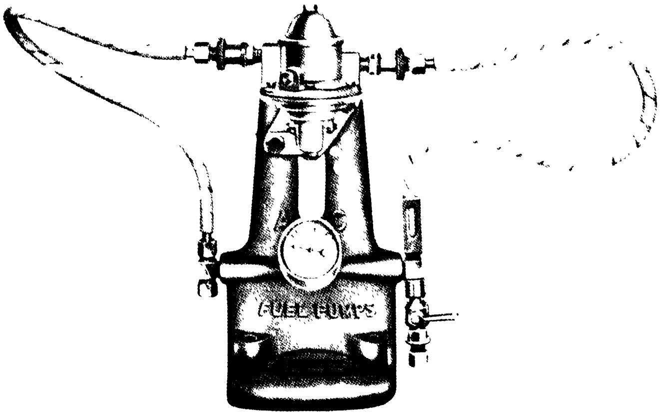

FIG. 66. The A C fuel-pump testing apparatus

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The test for a satisfactory pump is that the pressure reading should remain steady for several minutes; if the hand returns to zero in a relatively short time this indicates a leakage around the diaphragm or past the valves of the pump.

Another apparatus, of a somewhat similar type, known as the A.C. Petrol Pump Analyser, has been produced for making performance tests of fuel pumps under actual operating conditions, i.e. whilst the engine is running under its own power. Readings of petrol flow in unit time and petrol pressure on the delivery side are given by this apparatus. Vacuum tests can also be made, in order to indicate the existence of any obstruction in the petrol pipe line.

Oil Measurement.—The oil consumption of an internal combustion engine is usually measured by means of a float or depth gauge. The sump is filled to a pre-determined level at the commencement of the tests, observing its level with the engine warmed up. At the conclusion of the test, fresh (warm) oil is added until the original level is reached. The amount of oil used during the duration of the test is therefore the amount added. A convenient form of depth gauge is shown in Fig. 67. The fine point of the adjusting screw should just touch the surface of the oil when adjusting the level.

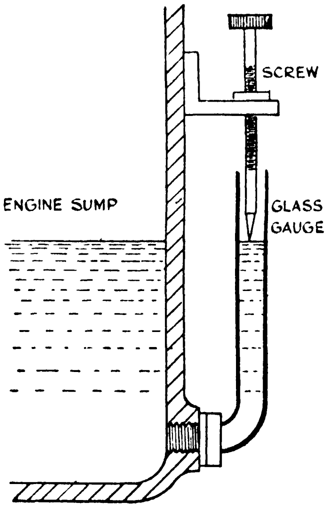

FIG. 67.—Showing one method of measuring oil consumption.

Oil Consumption.—The oil consumption in test records is usually expressed in "pounds of oil per B.H.P. hour."

The oil consumption of a modern water-cooled petrol engine is usually from 0·015 to 0·025 lb. per B.H.P. hour, the higher value corresponding to the maximum output. For air-cooled engines of recent design, e.g. motor-cycle and aircraft types using the dry sump method of lubrication, the corresponding values vary from about 0·025 to 0·045 lb. per B.H.P. hour.

Exhaust Gas Analysis.—The testing of the composition of the exhaust gases provides a useful check upon the mixture strength, and, therefore, upon carburation. For research purposes in connection with combustion and carburation, the results of exhaust gas analyses provide some useful information.

Although, strictly speaking, the volumetric analysis of a mixture of gases is the chemists' concern, yet, as we shall show, with the

aid of specially-designed apparatus and certain instructions it is possible for anyone of average intelligence and practical unacquaintance with chemistry to carry out a sufficiently accurate analysis. For less accurate purposes the direct reading type of exhaust gas analyser can be used.

The importance of being able to check the mixture strengths on commercial vehicles on the road, on isolated engines, aircraft engines in flight, and in other similar circumstances, in which it is not possible to measure fuel and air quantities, will be realized. In this manner, also, it is possible for owners of large fleets of motor vehicles to keep an accurate check upon their fuel bills, without having to resort to the inconvenient expedient of fitting every vehicle with fuel-measuring apparatus.

It has been stated that a certain commercial motor-vehicle company in America was able, as a result of periodic exhaust gas analysis tests, to effect a saving of from 15 to 20 per cent. in the fuel costs.

Exhaust Gas Composition.—As a result of chemical analysis of petrols, and of the exhaust gases, it is now known how the composition of the exhaust gases depends upon the mixture strength, and further, it is possible, knowing the exhaust gas analysis for any given running conditions, to at once deduce the corresponding mixture strength, and, therefore, to ascertain whether the engine has been running under the best carburation conditions.

It has been shown that the ordinary fuels used in automobile and aircraft engines consist of carbon, hydrogen, and (in some cases, as with alcohols) of oxygen, whilst the atmospheric air consists principally of nitrogen (76·8 per cent. by weight) and oxygen (23·2 per cent. by weight).

When a hydrocarbon is completely burnt with air, the principal products are nitrogen, water, and carbon dioxide. The theoretical weights and volumes of these products can be calculated by the method previously given.[1]

For example, when nearly pure petrol, or hexane, C_6H_{14}, is mixed with air and burnt in the proportions of 1 part petrol to 15·2 parts of air by weight, the theoretical composition of the products (neglecting the water formed, which is regarded as condensed) is

Nitrogen (N_2) = 85·7 volumes.
Carbon dioxide (CO_2) = 14·3 ,,

In the case of an actual engine, the range of mixtures used is such that complete combustion occurs only at one value, or ratio, the other mixtures being weaker (in fuel) or richer than this.

For weaker mixtures there is an excess of air, so that it will be

[1] *Vide* p. 8.

evident that the exhaust products will contain free oxygen as well as carbon dioxide. On the other hand, in the case of rich mixtures there will not be enough air to burn the whole of the fuel, so that partial combustion, or carbon separation, occurs. When a hydrocarbon is only partially burnt, *carbon monoxide* (CO) is formed as well as CO_2 and hydrogen, H_2. Experiment shows also that another gas, namely, *methane*, CH_4, is formed in small amounts in the case of rich mixtures, together with traces of a hydrocarbon known as aldehyde ($C_nH_{2n}O$). Summarizing these results we have the following gaseous products :—

1. **Weak Mixtures.**—CO_2, O_2, and N_2.
2. **Perfect Combustion Mixture.**—CO_2 and N_2.
3. **Rich Mixtures.**—CO, CO_2, N_2, H_2, CH_4, and aldehyde (traces).
4. **Very Rich Mixtures.**—Items as in (3), with addition of free carbon, in the form of sooty deposits.

It will be observed that CO and O_2 do not occur together in any mixture.

The following is a typical exhaust gas analysis result for a mixture of 14·75 parts air to 1 part petrol of 0·720 sp. gr. at 15° C. This mixture gave the nearest approach to complete combustion :—

Carbon dioxide	CO_2	=	13·3	per cent.	by volume.
Carbon monoxide	CO	=	0·5	,,	,,
Oxygen	O_2	=	0·5	,,	,,
Methane	CH_4	=	0·06	,,	,,
Hydrogen	H_2	=	0·18	,,	,,
Nitrogen	N_2	=	85·46	,,	,,
Aldehyde	$C_nH_{2n}O$	=	traces	,,	,,
			100·00	,,	,,

This is about the only mixture strength at which very small quantities of oxygen and carbon monoxide occur together.

Fig. 68 illustrates the results of a very large number of exhaust gas analyses, in the case of a petrol engine employing petrol of 0·720 sp. gr. The curves given show the volumetric percentages of the gases named, for various mixture strengths over the whole working range. The nitrogen curve is not given, but it may be obtained readily by the difference method. Although these curves vary to a small extent with the composition of the petrol, they may be taken as being sufficiently representative for most practical purposes.

The following general deductions can be made from Fig. 68 :—

1. That for complete combustion (at a mixture strength of 15·2) there is the maximum percentage of CO_2 present, for any

mixture; in this case 13·3 per cent. Further, only a very small percentage of CO and O_2 are present together; this is probably due (*a*) to air leakage into the exhaust pipe, past the exhaust valve stem, etc.; and (*b*) to occasional misfiring of one or more charges.

2. That for weak mixtures (in petrol), CO_2 and O_2 are present. The weaker the mixture the less the CO_2 and the more the O_2 present.

3. That for rich mixtures, CO and CO_2 are present. The percentage of CO increases, and of CO_2 decreases as the mixture enrichens.

4. The percentage of CO present in the exhaust gases varies as the fuel wasted, and is, therefore, an indicator of the fuel economy or otherwise.

For each fuel, or fuel mixture, it is necessary to construct an exhaust gas analysis chart [1] similar to that shown in Fig. 68. It is unnecessary here to enter further into the experimental and

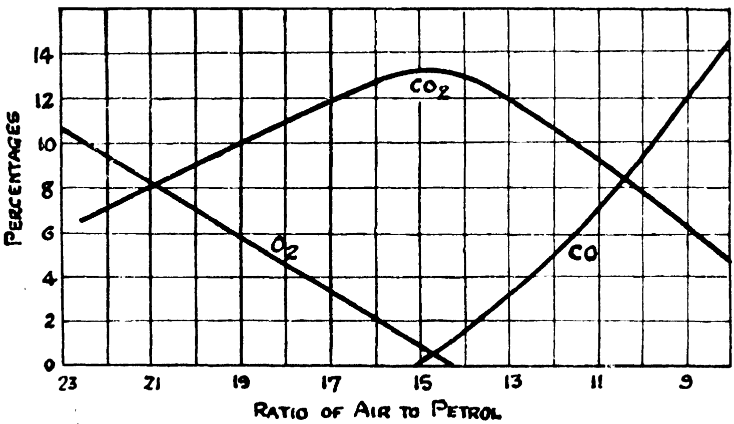

FIG. 68.—Exhaust gas analysis curves.

theoretical aspects of the subjects of combustion and exhaust gases, but to those desiring to pursue these subjects, the footnote references [2] on this page are given.

Having seen how the exhaust gas compositions vary for different mixture-strengths, and having also shown how a typical chart of compositions is constructed, it will be evident that by analysing the exhaust gases in any specific case, the equivalent mixture strength can at once be deduced, and combustion conditions studied.

Exhaust Gas Analysis Apparatus.—As previously mentioned, exhaust gas analysis with modern apparatus can be carried out

[1] Typical examples are given in "Benzole, Alcohol, etc., as Fuels for Internal Combustion Engines," W. Watson, F.R.S., *Proc. Inst. Autom. Engrs.*, 1914-15.

[2] "The Thermal and Combustion Efficiency of a Four Cylinder Petrol Motor," W. Watson, F.R.S., *Proc. Inst. Autom. Engrs.*, 1908-9.

"Automobile and Aircraft Engines in Theory and Experiment," A. W. Judge (Sir Isaac Pitman & Sons).

as a fairly simple routine process, without the necessity for much chemical knowledge. The most convenient apparatus for the purpose is probably the Macfarlane-Caldwell type, supplied by Messrs. Baird & Tatlock, London, which is shown illustrated in Fig. 69. The principle of the use of the apparatus consists in bringing a measured volume of the exhaust gases successively into contact with different solutions, each of which has the power of absorbing one of the gaseous constituents present. After each absorption the

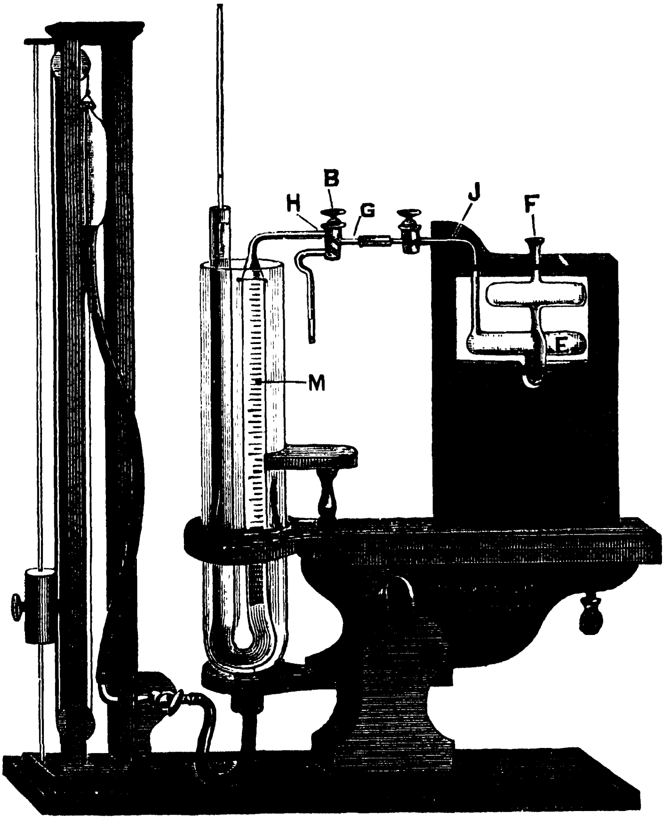

FIG. 69.—The Macfarlane-Caldwell gas analysis apparatus.

volume of the remaining gases is measured (always under atmospheric conditions and at a constant temperature, which is ensured by a water-bath), and the volume of gas absorbed is obtained by the method of differences.

The following are the solutions employed :—

1. *Caustic Potash solution*, consisting of 50 grammes of caustic potash dissolved in 100 c.c. of water. This solution absorbs the CO_2.

2. *Alkaline Pyrogallol* (5 grammes of pyrogallol and 50 grammes of caustic potash dissolved in 100 c.c. of water). This solution absorbs the O_2.

3. *Acid Cuprous solution* (used). This is prepared by dissolving 36 grammes of cuprous chloride in a mixture of 100 c.c. of water and 150 c.c. of concentrated hydrochloric acid. The solution, when not in use, is kept in a bottle half-filled with copper turnings to prevent oxidation. This solution is used for the absorption of the bulk of the CO present.

4. *Acid Cuprous solution* (fresh), prepared as above. This is used for absorbing the last portions of the CO.

The apparatus shown in Fig. 69 consists of the U-tube measuring gas burette M, which is placed in an outer glass vessel filled with water, to keep the temperature constant throughout a test.

There is a mercury container, shown on the left, connected with the lower part of M, or rather the stop-cock below M, by means of a flexible rubber tubing. This mercury reservoir can slide up and down vertically, and its purpose is to expel the gases in M into the absorption vessel E (by raising the reservoir, with both of the cocks shown, open), and to return the gases into M afterwards. A special stop-cock B enables M to communicate with E, or alternatively E to communicate with the lower capillary tube, which is open to the atmosphere. This latter connection enables the absorbing liquid in E to be forced through G and the lower passage in the cock B, thence through the atmospheric open tube. By blowing at the open end F, the capillary tube G can thus be filled, and the cock B turned, so that there is no loss of gas due to the capillary tubes.

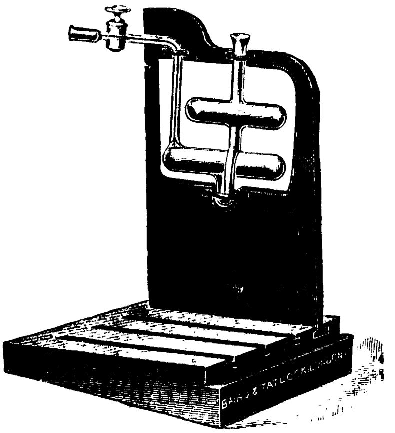

FIG. 70.—Absorption vessel on stand.

The wooden stand carrying the absorption vessel E (Fig. 70) slides in, and is detachable from, the main frame carrying the water vessel, so that different absorption vessels can be used in turn. Moreover, by taking out the lower right-hand pin on the stand the apparatus can be rocked about the other pin in order to facilitate the absorption of the gases by the solution in E.

The method of making an analysis is briefly as follows :—

The exhaust gases, having been collected by a method which will be explained later, in a glass vessel provided with capillary tubes and stop-cocks at each end, the glass vessel is connected by

a short length of rubber tubing to G. Previously to this, the vessel M has been cleared, and (by lowering the mercury reservoir) filled right up to the capillary tube G with mercury, all air being thus expelled. When the glass vessel is connected, its nearest stop-cock to B, and also B, and the stop-cock below M, are all opened, and the mercury reservoir raised until about 50 c.c. of exhaust gases are drawn into M. The final measurement of the volume, after allowing due time for temperature conditions to settle down, is made under atmospheric conditions by adjusting the mercury levels to the same height in each limb of the U-tube, of which M forms one.

Next the CO_2 absorption tube E is slid into place, coupled up at G, and the capillary tubing JG filled with solution as explained before. The exhaust gases are then forced into the lower vessel E; part of the caustic solution is thus displaced into the upper vessel above E, so that an effective gas seal is made. The stand of E having been tightened by means of the thumb-screw shown below on the right-hand side, the locking pin is withdrawn from the stand and the apparatus rocked for about 30 *seconds*, after which the remaining gases are passed back into M, the mercury levels adjusted, and the volume read off. The difference between the initial and final volumes represents the volume of CO_2 present; this can be expressed as a percentage of the initial volume. Similarly the O_2 can be absorbed by the pyrogallol solution. In this case it is necessary to rock for about 2 *minutes*. Then the CO is absorbed, firstly in the used cuprous chloride solution for 3 *minutes*, and secondly in the fresh cuprous chloride solution for 2 *minutes*. After each complete absorption the volume is measured.

As there is a possibility of a very small amount of CO_2 being set free by the acid cuprous liquid upon traces of the other reagents adhering to the glass, it is necessary—where great accuracy is required—to give a final absorption in the caustic solution. The *hydrogen* present can be measured by a special apparatus, but for practical purposes can be estimated with sufficient accuracy from the relation.

$$\text{Percentage of } H_2 = 0{\cdot}36 \text{ (percentage of CO)}.$$

Similarly, the *methane* is calculable from the empirical relation.

$$\text{Percentage of } CH_4 = 0{\cdot}12 \text{ (percentage of CO)}.$$

By exploding the gases remaining (after absorption of the CO_2, O_2 and CO) in a special piece of apparatus, with oxygen and measuring the CO_2 formed, the presence of any other unburnt hydrocarbon can be tested for.[1]

For commercial purposes, if a gas analysis chart such as that shown in Fig. 68 is available, it is usually only necessary to measure

[1] The author's tests in this connection have revealed only traces of hydrocarbon.

the CO_2, O_2, and CO, in order to deduce mixture-strengths ; in fact the measurement of any *one* of these is sufficient for carburation testing purposes.

Road Vehicle Engine Tests.—The gas analysis apparatus used in the research department of Messrs. Associated Equipment Company, Ltd., Southall, Middlesex, in connection with commercial vehicle engines is shown in Fig. 71. The important parts are lettered.

The Orsat apparatus illustrated has three containers, A, B, and C, containing the following solutions :—

(A) Ammoniacal copper chloride for absorbing CO.
(B) Alkali pyrogallic acid for absorbing oxygen.
(C) Caustic potash for absorbing carbon dioxide.

The graduated vessel on the right marked E is a burette into which is drained by water displacements a certain quantity of

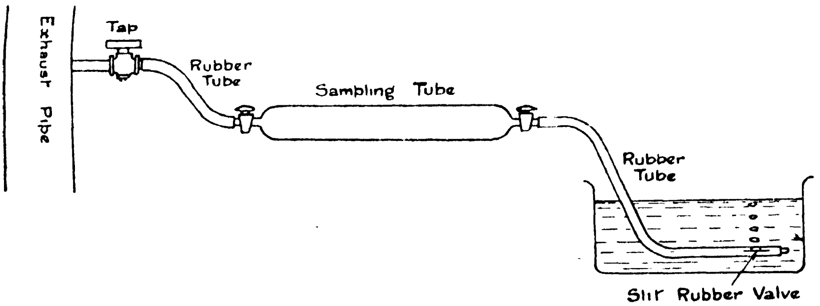

FIG. 73.—Method of collecting sample of exhaust gases.

exhaust gas. It is then possible by moving the vessel containing water, which is marked F, up or down in relation to the burette E to displace this exhaust gas back through the absorption vessels, the cock G being closed during progress.

The procedure when in use is that cock G opens to draw in a certain quantity of exhaust gas. Cock G is then closed. Cock H opens and the gas caused passes through the absorption vessel A, which will remove the carbon monoxide. The remaining gas is then drained back into the measuring burette and its volume checked. By this means it is possible to state the volume of carbon monoxide contained in the sample undergoing analysis and by a similar process in the case of oxygen and carbon dioxide.

Method of Collecting Exhaust Gases.—The exhaust gases are usually collected in a glass sampling vessel of about 300 c.c. capacity, having capillary tube ends and provided with stop-cocks at each end.

The gases are allowed to blow through from the exhaust pipe as shown. A piece of rubber tubing is connected to the outer end

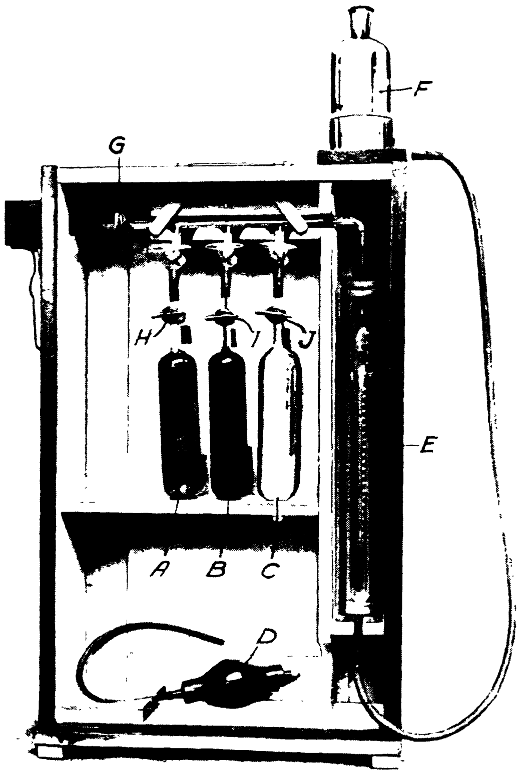

FIG. 71.—Exhaust gas analysis apparatus used for A.E.C. engine tests.

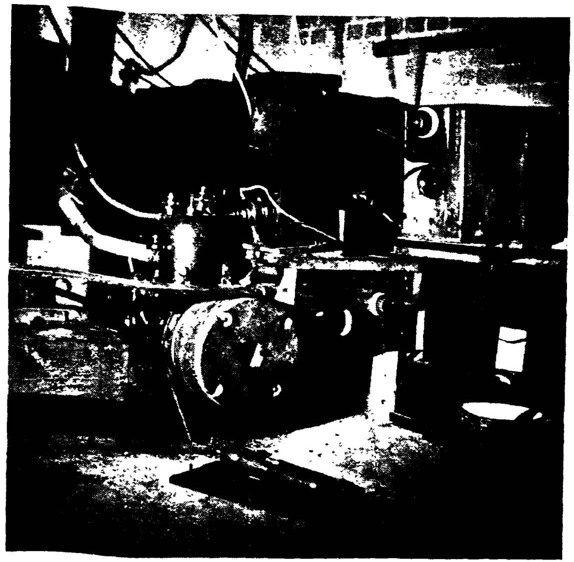

FIG. 72.—Showing method of taking exhaust gas sample from a two-stroke engine. The optical indicator and its arc lamp illuminator are seen above, to the right of the engine.

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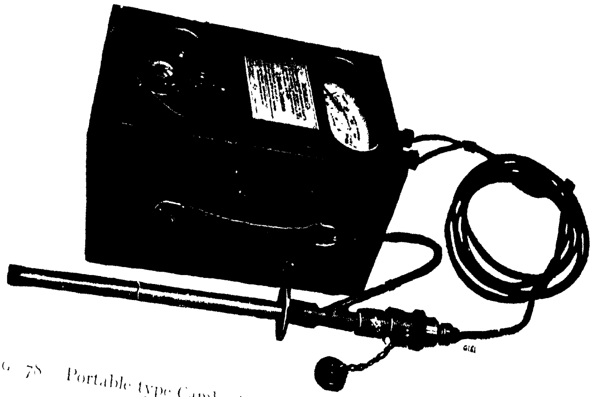

FIG. 78. Portable type Cambridge CO_2 and exhaust temperature indicator.
[See page 124.

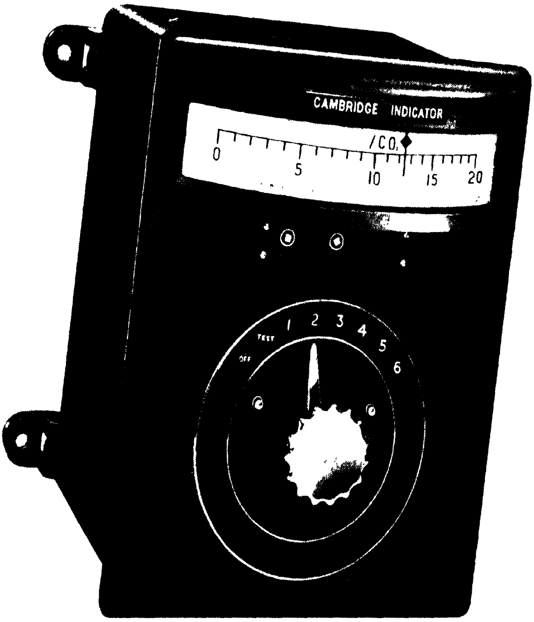

FIG. 79. A six-point carbon-dioxide indicator.
[See page 125.

of the sampling tube, and led under water. The end of this rubber tubing is plugged with a piece of glass rod, or other suitable material, and a small slit is made in the tubing, to act as a non-return valve.

The gases should be allowed from 3 to 4 minutes to blow through, in order to displace the air inside.

Alternatively a quicker method is to fill the sampling tube and its connection to the exhaust pipe cock with mercury, and, by allowing the mercury to flow out slowly, with the sampling tube vertical, the exhaust gases can be drawn into the latter.

Sampling Valve for Cylinder Heads.—In connection with an investigation into the fuel distribution in the individual cylinders of a multi-cylinder car engine by the Ethyl Gasoline Corporation, U.S.A.[1] the method employed was to take samples of the exhaust gases from the combustion chambers instead of the exhaust manifold, so that the equivalent mixture strengths could be estimated from the analyses of the samples. The method employed was to use a special design of combination sparking plug and gas-sampling valve (Fig. 74). The latter consisted of a small solenoid valve between the cylinder and gas sampling tube and was controlled by an auxiliary contact-breaker mechanism mounted above the ignition distributor of the engine. At the instant the points of the auxiliary breaker close, a condenser in the circuit discharges a relatively heavy current through the solenoid coil and causes the valve to open momentarily. At other times the valve is held closed by a coil spring. The complete apparatus is shown diagrammatically in Fig. 75. When a sample is to be obtained, the vacuum pump, driven by an electric motor, evacuates the gas-sampling tube to an absolute pressure of about 5 mm. of mercury. The car is accelerated under the conditions prevailing in the road knock test, and the valve allowed to operate only within the 5 m.p.h. speed range of maximum knock intensity. From six to eight repetitions are required to fill the sampling tube to atmospheric pressure. The procedure is then repeated for the other cylinders. The results obtained for a certain

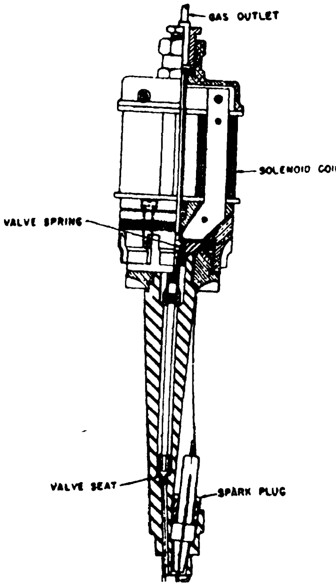

FIG. 74.—Gas sampling valve and sparking plug unit.

[1] "The Effects of Carburation and Manifolding on the Relative Anti-Knock Value of Fuels in Multi-Cylinder Engines," E. Bartholomew, H. Chalk and B. Brewster, *Journ. Soc. Automotive Engrs.*, 1938-9.

eight-cylinder engine with dual carburettor indicated a mixture ratio variation between the limits of 16·01 to 1 in cylinder No. 2, and 11·56 to 1 in cylinder No. 6; the average ratio for the eight cylinders was 13·6 to 1.

Carbon Dioxide and Oxygen Indicators and Recorders.—The method of exhaust gas analysis described in the preceding pages gives as complete an analysis as ordinary test and research considerations require. In the case of engines undergoing long duration tests, and for the purpose of checking the fuel-air mixtures of commercial engines which run under more or less constant conditions, as, for example, in the case of stationary and marine types

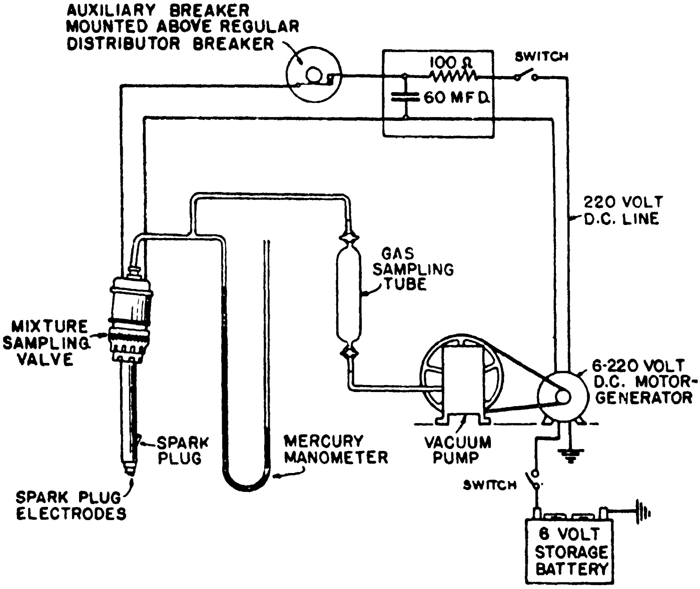

FIG. 75.—Apparatus for gas analysis from individual cylinders of car engine.

driving machinery, generators, etc., it is often advantageous to record automatically the principal contents of the exhaust products.

It has been shown that if, say, the percentage of CO_2 in the gases is measured, then for any given fuel an empirical relation exists between this value and the mixture strength; by measuring the one, therefore, we have a check on the other.[1] There is a number of automatic CO_2 and O_2 recorders upon the market, designed primarily for recording the percentages of these gases in the flue gases of boiler and other furnaces; with a certain amount of adaptation these are applicable to internal combustion engines.

In internal combustion engine work for the best thermal efficiencies the exhaust gas compositions should correspond to

[1] It is necessary to ascertain whether there is any oxygen present, in order to determine whether the CO_2 measured corresponds to a weak mixture or a rich one.

rather weaker mixtures than for complete combustion, namely, by about 15 per cent., so that the percentage of CO_2 in the exhaust gases should be about 12 in the case of petrol. For complete combustion it will be about 13·5 per cent.

The Cambridge CO_2 Indicator and Recorder.—The principle of the method used in this indicator is based upon measurements of the thermal conductivity of the exhaust gases, there being a fixed relation between this quantity and the amount of CO_2 in the gases. If a constant current is passed through a platinum wire placed in the gases the temperature of the wire will rise until a point of equilibrium is attained such that the electrical energy supplied to the wire equals the thermal energy lost by conduction through the gases. The temperature of the wire, therefore, depends upon the thermal conductivity of the gases. In the Cambridge electrical CO_2 indicator four identical platinum wire spirals are enclosed in separate cells, E_1, E_2, E_3, E_4, in a solid metal block, as shown diagrammatically in Fig. 76. Each spiral forms one arm of a Wheatstone Bridge circuit. A definite electric current is allowed to flow through the Bridge, thereby causing the spirals to become heated and to lose heat to the walls of the cells. If two gases having different thermal conductivities are introduced, one into two of the cells (say E_1 and E_3), and the other gas into the other two cells (E_2 and E_4), the spirals E_1 and E_3 will cool at a different rate from E_2 and E_4, and will therefore be maintained at a different temperature. The consequent difference in the electrical resistance of the spirals will throw the Wheatstone Bridge out of balance, causing a deflection of the galvanometer G, the extent of which depends upon the difference in the conductivity of the two gases. The construction is such that changes in the temperature of the metal block affect both sides of the bridge equally. If, therefore, the cells E_2 and E_4 contain a pure gas and the cells E_1 and E_3 the same gas mixed with some other constituent, the extent of the deflection will be an indication of the amount of the second gas present, and the galvanometer can be calibrated to show directly the percentage composition of the mixture.

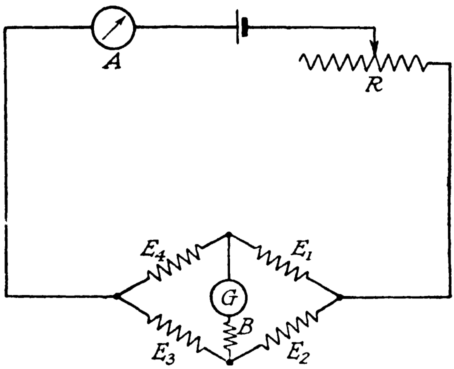

FIG. 76.

Variations in the proportion of nitrogen and oxygen have very little effect on the readings, as these two gases have practically the same thermal conductivity. The effect of water vapour is compen-

sated by keeping the gases in all the cells saturated. In practice two of the cells are filled with air, saturated with water vapour, and the other two are exposed to the gas under test, which is also saturated with water vapour at the same temperature. The difference in conductivity of the gases in the cells then depends solely on the percentage of carbon dioxide in the gas.

Fig. 77 shows the original type of metal block containing the cells, screwed into the pipe line. In this diagram M is a removable cleaning box containing glass wool mixed with iron borings, to protect the sensitive elements against soot deposit and sulphuric acid. The cleaned gases only flow past the meter and diffuse into the exposed cell; there is no appreciable lag due to this diffusion or to the filter M. L is a small brass chamber containing water to keep the sealed air cell saturated with moisture, whilst P is a screwed gland containing the four-way cable, from the meter to the indicator or recorder.

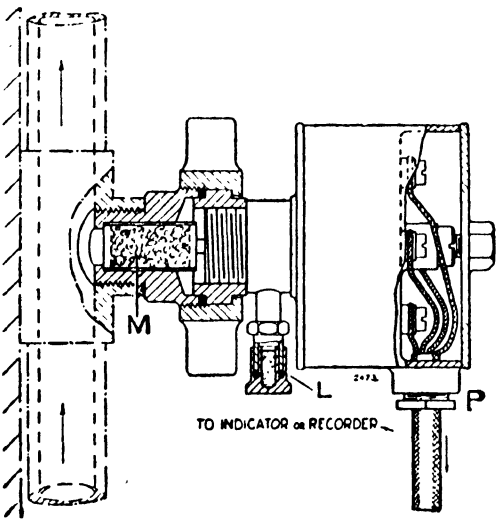

Fig. 77.

For flue gases, it is necessary to employ a separate soot-filter and aspirator of the water type.

Measurements of Carbon Monoxide.—For measuring the percentage of carbon monoxide and other combustible gases, the exhaust or flue gases are caused to pass through a small electric furnace where they are completely burnt, and a differential katharometer is used to measure the *increase* in CO_2 content, which depends upon the amounts of combustible gases originally present.

A typical CO_2 measuring outfit consists of an intake pipe and soot filter for sampling the exhaust or flue gases, connected by a pipe-line to the metering unit, which comprises the CO_2 meter proper, a bubbler for washing and saturating the gas and an aspirator for drawing the gas sample from the exhaust pipe or flue. The metering unit is connected electrically to an indicator or recorder and also to a source of electric current. For current control the galvanometer of the indicator or recorder is used as an ammeter thus avoiding the use of a separate instrument for this purpose.

The Cambridge CO_2 indicator is available in the single pattern laboratory, and also portable models. It is also made as a multiple

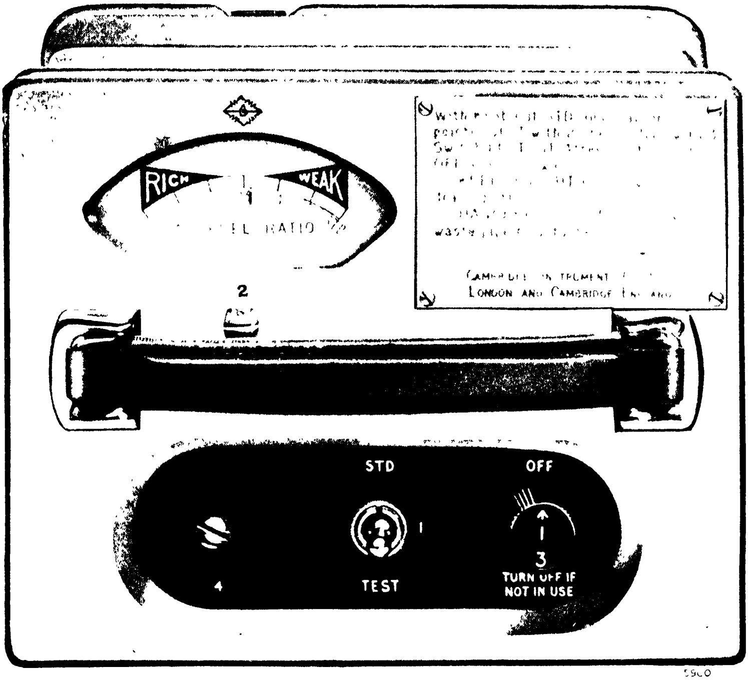

FIG. 85. The Cambridge exhaust gas tester

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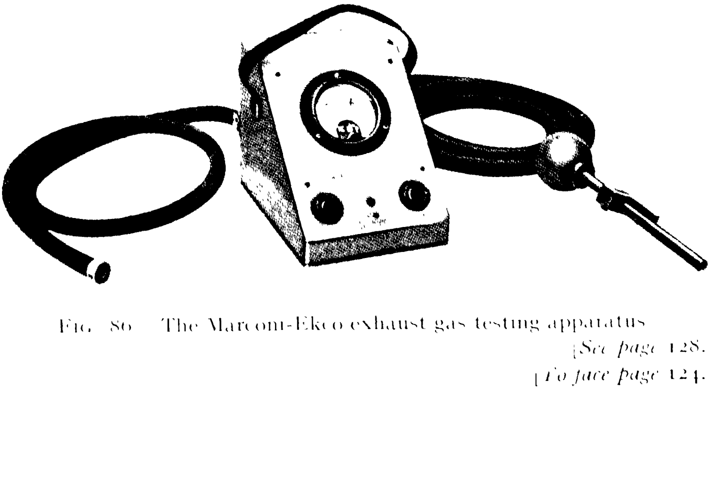

FIG. 86. The Marconi-Ekco exhaust gas testing apparatus

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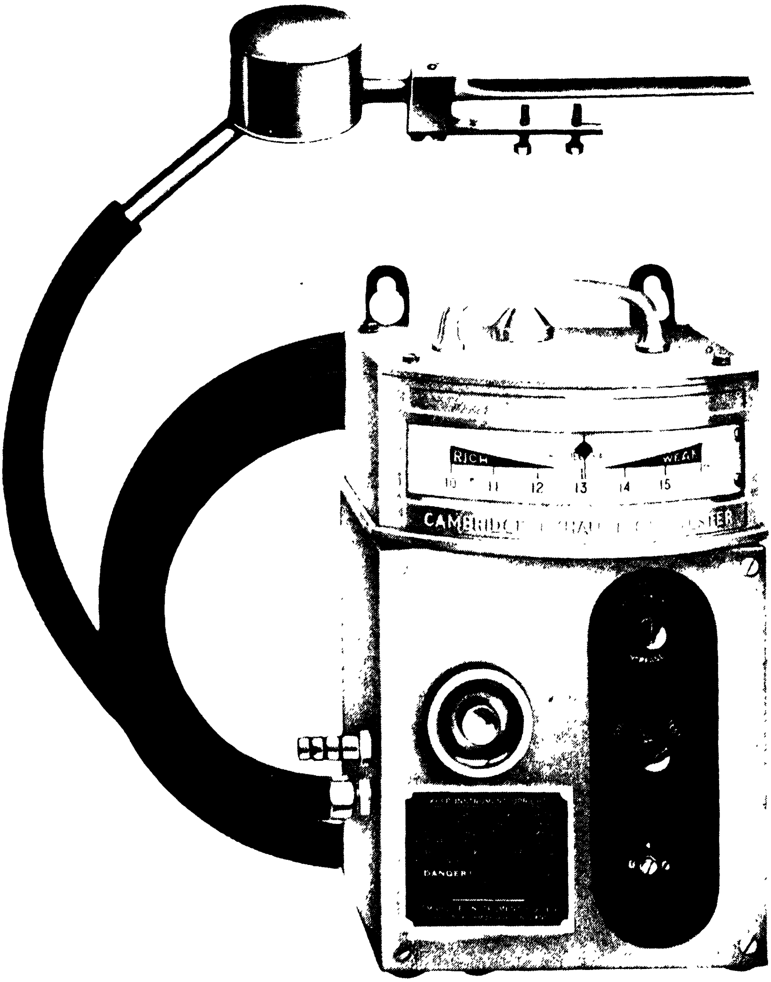

FIG. 81 Portable model Cambridge exhaust gas tester.

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instrument for indicating the percentage of CO_2 in six different exhaust manifolds or flue ducts (Fig. 79). For continuous readings a special type of recording instrument, with a 25-hour drum chart, driven by clock or electrically, is available for continuous recordings of one to six different stations.

The Cambridge Exhaust Gas Tester.—This instrument, which is available for both test house and road test purposes, operates upon a similar principle to that of the CO_2 indicator previously described. It utilizes the fixed relation that exists between the thermal conductivity of exhaust gases and the air-fuel ratio, so that the dial of the instrument is graduated to read air-fuel ratios over a range of 10 : 1 to 15 : 1 (Fig. 80). The maximum power ratio is taken as being 12·5 : 1 to 13·5 : 1.

The complete instrument is fitted in a cast aluminium case and is dust and water-proof. Two dry cells which furnish the current for operation are placed in a separate compartment for ready access purposes.

An important feature of the instrument is the unique method of suspending the moving system of the air-fuel indicator. The usual methods of mounting sensitive galvanometer systems are either to support the moving element with pivots, or to suspend it between fine tightly-stretched wires. Pivots may soon become blunt from vibrations and road shocks, and the movement of the coil becomes sluggish and its indications inaccurate. On the other hand, a suspended movement is susceptible to changes in level, and when used in a moving car the pointer may swing and jump so violently as to be unreadable.

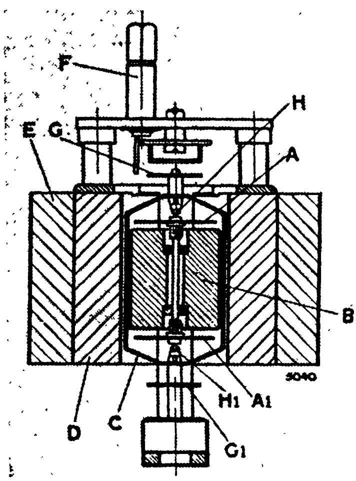

FIG. 82.—Cambridge magnetically cushioned movement. A, A1, iron discs magnetically suspended. B, iron core. C, coil. E, magnet. F, zero adjustment post. G, G1, control springs. H, H1, pivots.

The Cambridge magnetically cushioned movement, shown in Fig. 82, is so designed as to absorb the shocks of road testing without damage to the pivot point. The coil action is so controlled that the pointer remains steady and readable under all conditions encountered. The coil assembly, instead of being called upon to absorb shocks suddenly, floats in a magnetic field which cushions road shocks and protects the pivot from injury. This movement is very stable, rugged, and makes continuous exhaust testing practicable on the road.

This instrument is provided with a sampling hose 10 feet long,

and a tail pipe with fittings that enable it to be held in position on the exhaust tube. A special chamber collects the condensate and thereby prevents water from entering the instrument.

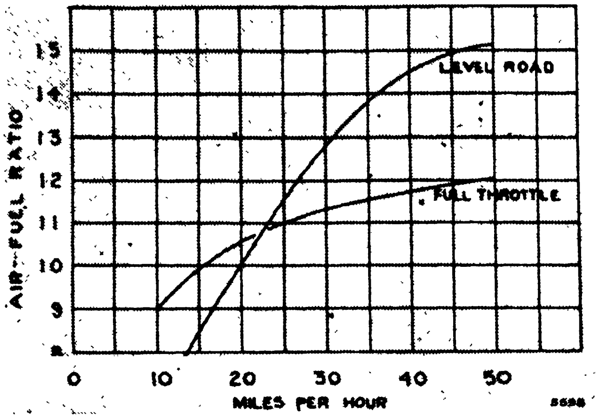

FIG. 83.—Results of tests before adjustment of carburettor.

Figs. 83 and 84 show the results of tests made with the Cambridge Exhaust Gas Tester before and after adjusting the carburettor on a motor-car. The results of the adjustments made with the use

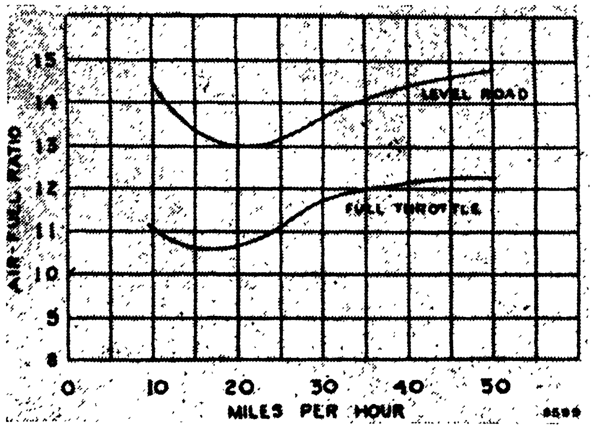

FIG. 84.—Results of tests after carburettor adjustment.

of the exhaust gas tester indicate a substantial improvement in the mixture regulation and also the provision of a richer mixture for acceleration purposes at the lower, intermediate, and maximum road speeds.

The Arkon CO_2 and O_2 Recorder.—This apparatus, although intended primarily for flue gas analysis, is applicable to exhaust gas analysis. It gives continuous records of the percentages of CO_2 and O_2.

Fig. 85 illustrates the principle of the CO_2 recorder, in which a charge of flue gas is sucked into the instrument, whence the CO_2 is absorbed in a vessel containing caustic potash. A water aspirator is employed to draw the charge in.

Referring to the illustration, clean water falls into the clean tank 10, and overflows through pipe 13 so as to keep a constant level in the tank. Aspirator 11, which passes through the tank, is perforated to allow water to flow from the tank into it. As this water falls down the pipe it induces a flow of the gases through the gas

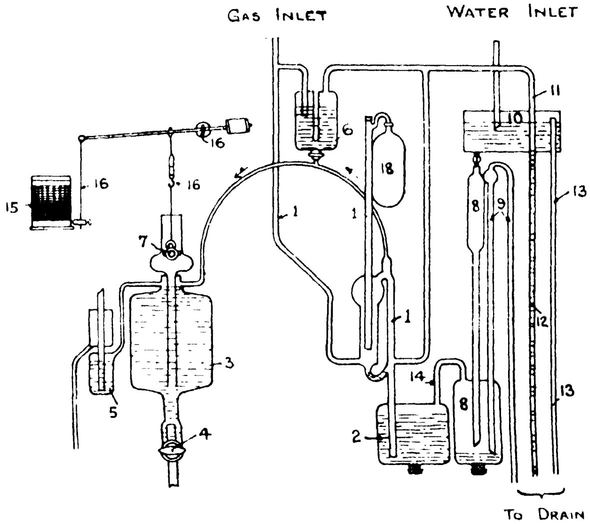

FIG. 85.—Illustrating the principle of the Arkon CO_2 recorder.

burette 1, which is connected with the sampling pipes from the exhaust or flue pipes. Another jet of water falls through tube 8 into the lower vessel 8, and, as this fills, the air is driven out on to the surface of the liquid in the "sealing liquid" vessel 2. The sealing liquid is thereby forced up into 1, first closing the left- and right-hand branches, and then, rising higher, drives the measured gas sample (100 c.c.) through the capillary tubing and on to the surface of the caustic potash in the absorption vessel 3.

Any CO_2 contained in the gas sample is absorbed, and the remainder forces an equal volume of solution up the central tube until it lifts the glass float 7, and pen gear 16. The greater the absorption of CO_2 the smaller the travel of the float 7. The percentage of CO_2 is, therefore, shown by the *unwritten* portion of the chart. At the end of the gas stroke, water, which has been rising

in the syphon 9, syphons over and discharges the water from the lower vessel 8. This allows the liquids in the "sealing liquid" vessel 2 and absorption vessel 3 to return to their normal position. To ensure a constant gas-flow and eliminate "time-lag" the glycerine seal vessel 6 is provided so that the gas can "short-circuit" across the aspirator 11, when the left and right branches of the vessel 1 are sealed by the rising liquid. The overflow vessel 5 is provided to maintain an exact volume of caustic potash in 3 at all times.

A useful feature of this instrument is the continuous check provided by the level of the liquid in the graduated burette 1, on the recorder drum readings.

In the combined CO_2 and O_2 recorder there are actually two separate absorption devices, one as previously described, for the CO_2, and the other (using stick phosphorus as the absorbent medium) for the O_2. In the latter case the small pieces of phosphorus are separated by glass beads and immersed in water. When a measured gas sample is passed over to the absorption vessel it depresses this water, leaving the phosphorus free to absorb the oxygen in the gases. The displaced water travels up to the float and actuates the pen gear. The chart ranges provided for oxygen recorders are 0 to 10 per cent., 0 to 21 per cent., 0 to 50 per cent., and 0 to 100 per cent.; the second range is applicable to exhaust gases.

In this type of instrument it is necessary to replenish the absorbing media at regular intervals; this is a fairly simple operation, however, in the present design, since every part is accessible and readily dismantled and assembled.

Marconi-Ekco Exhaust Gas Tester.—A convenient form of exhaust gas testing equipment for giving direct readings of the air-fuel ratio ranging from 15 : 1 to 10 : 1 is shown in Fig. 86. It has a 3-inch scale moving coil type of indicator and operates off two small standard dry cells; the current consumption is very small. It can readily be used on test engines in the laboratory and also on actual motor vehicles at rest or on the road. The instrument is compact, its dimensions being $10\frac{1}{2} \times 8 \times 7\frac{1}{2}$ inches; the total weight is 16 lb.

The Marconi-Ekco Exhaust Tester, illustrated in Fig. 86 (facing page 125), utilizes the principle of the variation of thermal conductivity with change in CO and CO_2 content in the exhaust gases.

Thermal conductivity is measured by means of a Wheatstone bridge consisting of four platinum wire spirals, two in air and two in the exhaust gas. The bridge unit is mounted in a heavy metal block to equalize the ambient temperature of the arms. The exhaust gas reaches the test arms by diffusion upwards past baffle plates which reduce temperature and condense out water vapour. The reading is not materially affected by variations in gas temperature

and smoke particles are largely excluded because of their low mobility.

A special "quick-fit" nozzle is inserted into the end of the exhaust pipe and the sample of the exhaust passes along a flexible rubber pipe and through the instrument. Where the instrument is placed inside the car for road tests an additional flexible rubber pipe carries the exhaust from the instrument out through the car window.

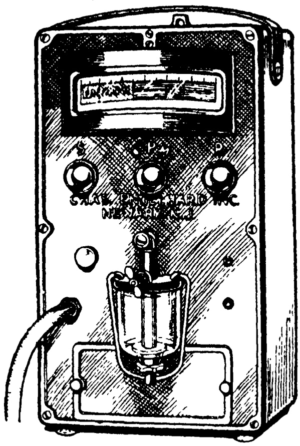

FIG. 87.—The Engelhard exhaust gas tester.

Another exhaust gas analyser of American origin for use in engine test laboratories and on actual motor vehicles is the Engelhard (Fig. 87); this gives continuous indications of mixture strength by measuring the proportion of CO_2 in the exhaust gases; it is based upon the principle of measuring the thermal conductivity of the exhaust gases. It is calibrated for a correct mixture strength equivalent to 15 parts air to 1 of petrol, and a corresponding percentage of CO_2 of 14·7. For a rich mixture of 12 to 1 the percentage of CO_2 falls to 9·4, whilst for a very rich one of 9 to 1 the value becomes further reduced to 5·7. The range of mixture readings on the Engelhard scale, however, is from 9 to 15, and a continuous reading is given whilst the engine is running.

Another apparatus of American origin, but available in this country for analysing exhaust gases of petrol and Diesel engines,

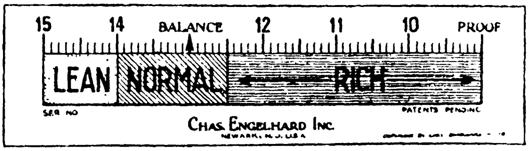

FIG. 88.—Showing scale of Engelhard exhaust gas tester.

is the Lantz-Phelps. It employs the Wheatstone bridge resistance method of measuring alterations in the electrical conductivity of the exhaust gases, the variations being indicated on a sensitive galvanometer showing on a suitable scale air-fuel readings from weak to rich. It is of the portable pattern and can be used for road tests. A 6 or 12 volt battery is required for the instrument.

CHAPTER IV

MEASUREMENT OF AIR SUPPLY

THE measurement of air quantity is an important one in carburettor and internal combustion engine research work. For this reason a brief account will be given of the more important methods applicable and in actual use.

At the outset it should be mentioned that the air flow is of a pulsating nature, in the present case, and for this reason many of the constant flow type of instruments are not directly applicable. If, however, this type is fitted either to a constant source of air supply to a gasometer or reservoir, from which the engine draws its supply, or, alternatively, is fitted in a position sufficiently remote from the source of pulsation, it can be used satisfactorily. Absolute measurement of air quantity involves a knowledge of its density, which necessitates measurements of temperature, pressure, and moisture content.

There are many available methods of measuring air quantity which are based upon the known physical properties of gases. Of these, the following will be described in the present chapter :—

1. The Throttle-Plate Method.
2. The Pitot, Flow-Nozzle, or Orifice Methods.
3. The Electric Air-Flowmeter Method.
4. The Direct Air Measurement Method.

1. **The Throttle-Plate Method.**—The principle of this method is to draw the air supply to the engine through a box or reservoir provided at its open (to the atmosphere) end with a circular orifice in a thin partition. The difference of pressure between the outside and the inside of the box, or, strictly speaking, on the two sides of the orifice, together with a knowledge of the barometer height, air temperature, and humidity, enables the quantity of air flowing in unit time to be computed.

This method was studied, in the case of steady flows, by A. O. Muller,[1] R. J. Durley,[2] W. E. Dalby,[3] Weisbach,[4] and others, but, unfortunately, their results disregarded the effects of pressure variations. In 1912, Messrs. Watson and Schofield [5] carried out

[1] *Zeitschrift des Vereines deutscher Ingenieure*, February, 1908.
[2] *Trans. Amer. Soc. of Mech. Engineers*, 1906, vol. 27.
[3] *Engineering*, September 9th, 1910.
[4] *Der Civilingenieur*, 1859, vol. 5, p. 546.
[5] "On the Measurement of the Air Supply to Internal Combustion Engines by Means of a Throttle Plate," Prof. W. Watson, F.R.S., and H. Schofield, A.R.C.S., *Proc. Inst. of Mech. Engrs.*, May, 1912.

a complete series of experiments on the measurement of air supply to internal combustion engines by the throttle-plate method, and were able to formulate a practical method for its application.

When a difference of pressure equal to p centimetres of water exists on opposite sides of a circular orifice of area F square centimetres, the volume V, in cubic centimetres, of air which will pass in t seconds is given by

$$V = atF\sqrt{\frac{2gp}{\rho}}$$

where ρ is the air density in grammes per cubic centimetre, and a is a coefficient the value of which varies with the pressure difference p, the area F, and also with the size of box used.

Messrs. Watson and Schofield determined the values of the coefficient values of p, expressing their results in the form of a series of tables of correcting factors. These tables are reproduced in Appendix No. V.

The three air boxes used for these experiments measured $57 \times 25 \times 25$ inches, $38 \times 17 \times 17$ inches, and $28 \times 13 \times 13$ inches respectively, corresponding to capacities of 17·36, 4·66, and 1·53 cubic feet respectively.

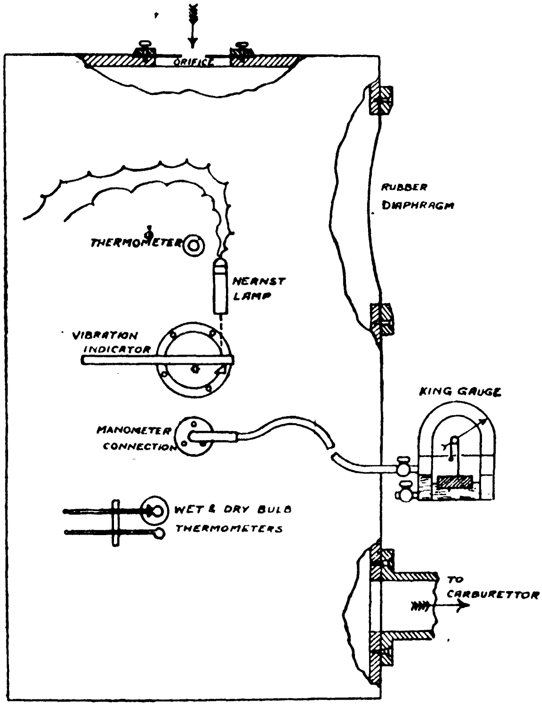

FIG. 89.—Illustrating general layout of throttle-plate air measurement apparatus.

The diaphragms were made from tin-plate of thickness 0·4 mm. (27 S.W.G.), and the orifice holes were turned in a lathe, and finished with a sharp edge. The diameters of the orifices used varied from ½ to 2 inches. The difference in pressure between the two sides of the orifice was measured with the modified form of King gauge shown in Fig. 90. Probably the more modern wind-channel pressure-difference gauges or manometers would be more suitable

for this purpose now, although it should be mentioned that readings to $\frac{1}{100}$ inch of water were obtainable with the gauge illustrated.

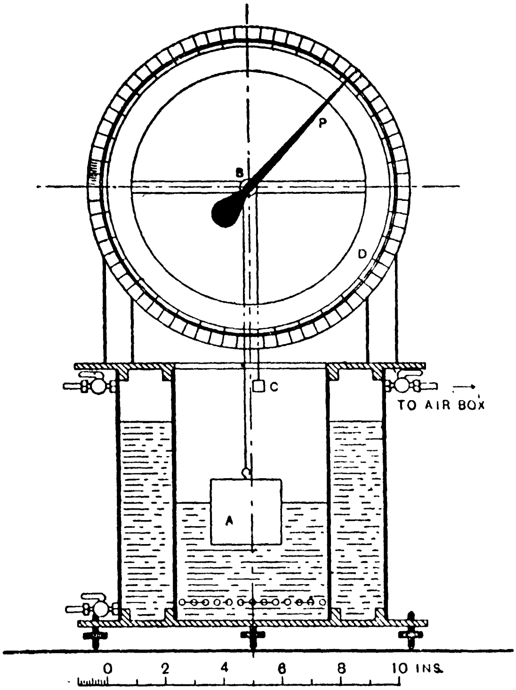

FIG. 90.—Modified King gauge.

The results of these experiments showed that for the same size of orifice (1 inch) and pressure difference, the coefficient a was practically constant (at 0·595) for the two larger boxes, but increased (to ·600) for the smallest box.

The effect of periodic pressure variations may, of course, be reduced by providing a sufficiently large box, but its dimensions might not be convenient for the larger types of automobile or air-craft engine. If the amplitude of the pressure variation is known, or measured, and its value be denoted by $a_1 p$, where p is the mean pressure difference, then the corresponding value of the coefficient a_m for pulsating flow is given by

$$a_m = a\left(1 - \frac{a_1^2}{16} - \frac{15a_1^4}{1024}\right)$$

where a is its value for steady flow (*vide* Fig. 91).

The paper in question contains tables of value of a_m for different values of a, and for different orifices, etc.

The value of the coefficient a, for the large box, when used for measuring the air supply for a four-cylinder engine, 85 mm. bore and 120 mm. stroke, was found to be independent of the existing pulsation.

Example of Method of Calculating Air Flow.—Referring to the Tables A to E, given in the Appendix, the following example will serve to make clear the method employed :—

Orifice diameter = $1\frac{3}{4}$ inches ; temperature of air = 20° C. (68° F.) ; barometric height = 754 mm. ; pressure difference on either side of the throttle-plate orifice = 0·800 inch water. Amplitude of pressure variation = 0·5 inch water.

$$\left(a_1 = \frac{0·5}{0·8} = 0·62\right).$$

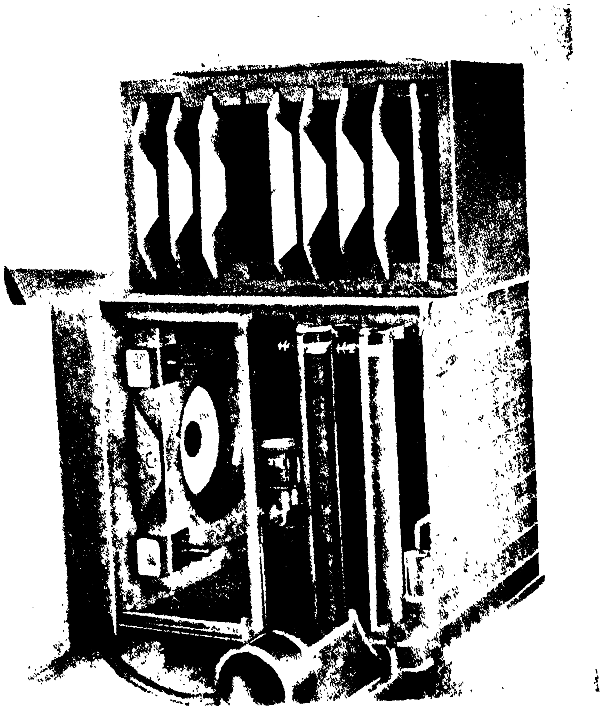

FIG. 92.—Bureau of Standards air-flow apparatus.

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From Sub-Table A it will be seen that 0·0500 lb. of dry air will pass through a $1\frac{3}{4}$-inch hole when there is a steady pressure difference of 1 inch of water, the temperature being 15° C. (59° F.), and barometric pressure 760 mm. (29·6 in.). This value must, therefore, be corrected for the influence of the existing factors by multiplying it by the following correction coefficients :—

Factor for	pressure difference	(Sub-Table B)	= 0·894
,,	temperature	(,, C)	= 0·992
,,	barometer	(,, D)	= 0·996
,,	change in coefficient	(,, E)	= 1·001
,,	variable pressure	(Fig. F)	= 0·974

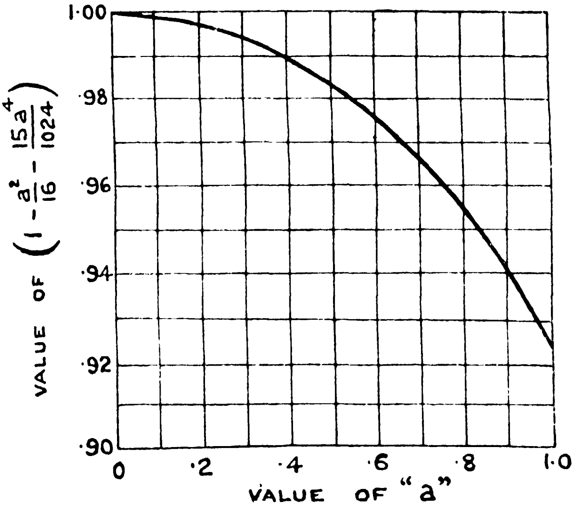

FIG. 91.

Hence the actual weight of dry air passing per second will be

$$W = 0{\cdot}0500 \times 0{\cdot}894 \times 0{\cdot}992 \times 0{\cdot}996 \times 1{\cdot}001 \times 0{\cdot}974 = 0{\cdot}0431 \text{ lb.}$$

Application of the Throttle-Plate Method.—With suitable precautions this method will give accurate results, namely, to within one-half of 1 per cent.

It is necessary to erect the measuring box in such a position that the throttle-plate flow is undisturbed by external air currents or draughts. In this respect it is of interest to note that the Bureau of Standards of America employed a honeycomb or grid approach passage for the purpose of protecting the lines of flow into the orifice from stray air currents.

Fig. 92 illustrates the arrangement adopted in the Bureau of Standard's carburettor testing plant ; it shows the protecting grid and spare orifice-diaphragms. In this case the diaphragms

were 0·057 inch thick, and the orifices ranged in size from 0·5 to 3·5 inches. The orifice plates seated upon a rectangular gasket of pure gum rubber of ¼-inch thickness to ensure air-tightness. The pressure drop across the orifice was read on the upper of two inclined manometers. This manometer was provided with divisions 1 mm. apart, so arranged that each division was equivalent to a vertical rise of 0·01 inch. This permitted of an accuracy of reading to within $\pm$ 0·0025 inch.

The air-box employed with the throttle-plate method must be perfectly rigid in construction and air-tight; also the pipes to the carburettor. A judicious application of Chatterton's compound to small leak sources will be found effective.

The box may be tested for air leaks by fitting a blank diaphragm in place of the orifice, sealing the outlet end, and pumping up to a definite pressure as registered by the manometer attached to the air-box. The time taken for this pressure to fall to one-half its original reading is noted; from this the rate of leakage can be computed. The places of leak can be found by filling the box with coal-gas, or by the soapy water test.

Experience has shown that all metal sheet zinc covering plates soldered all around at the joints is most effective.

For any given series of tests, say, on a limited range of engines, any given air-box and diaphragm may be calibrated in place by means of a suitable gas-holder, and thereafter only the air-box instrument readings taken. The diaphragm orifice should be well rounded (i.e. trumpet-shaped), since it is difficult to make and retain a true knife-edge.

2. **The Pitot, Flow-Nozzle, and Orifice Methods.**—This method has attained considerable commercial success for measuring the steady flow of air, steam, or gas. It possesses the advantages of cheapness, simplicity, and compactness, and, moreover, can be readily applied to recording and integrating meters.

It should, however, be pointed out that the method adopted, namely, that of measuring changes in velocity head, is only applicable, strictly speaking, to uniform flow, but as we have mentioned, it is not a difficult matter to provide a sufficiently large reservoir to damp out the usual fluctuations. As the number of cylinders per engine unit increases, so do these pulsations diminish in amplitude and in frequency.

With an air reservoir of about 250 times the cubical capacity of the engine, the pressure variations, in the case of a four-cylinder engine, will be found to be of comparative unimportance.

Messrs. The B.T.H. Co. employ three types of apparatus, namely: (*a*) The Pitot Tube; (*b*) The Nozzle Plug; and (*c*) The Orifice Tube. The former two apply to pipes of diameters above about 2 inches, and the latter to pipes of 2-inch diameter and below.

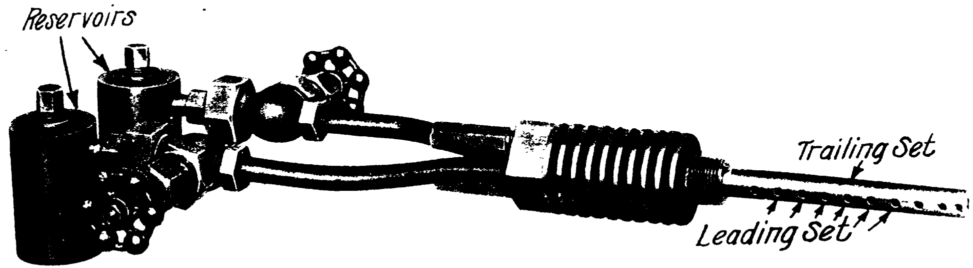

FIG. 94.—The B.T.H. nozzle plug.

[See page 135.

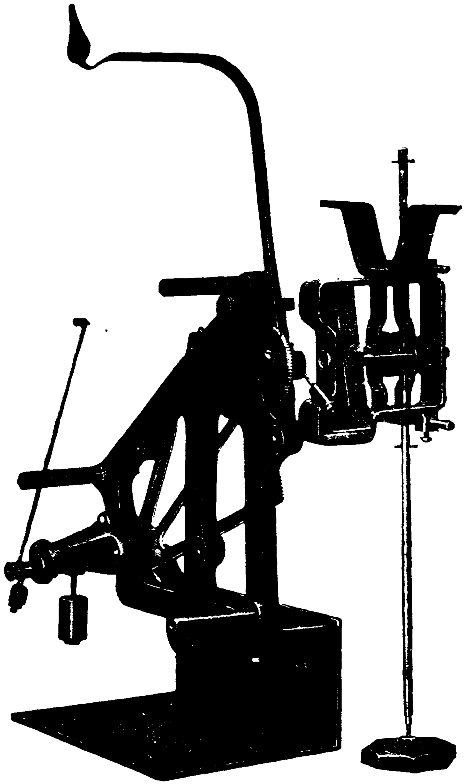

FIG. 95.—Mechanism of the indicating recorder

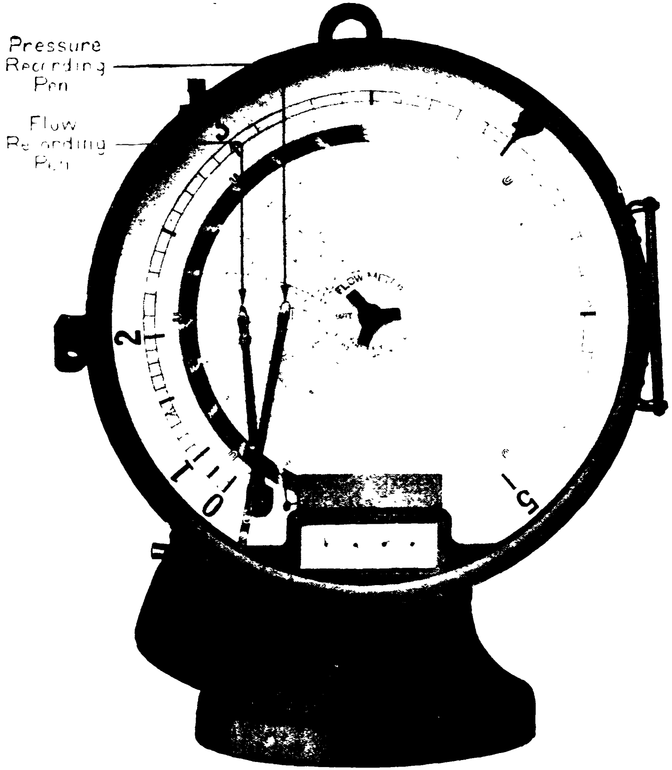

FIG. 96.—The B.T.H. air-flow meter indicator recorder.

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The principle of the first method is illustrated diagrammatically in Fig. 93. It consists of a special shape of pipe inserted in the air main, so that its two openings lie in the path of the air. The leading opening faces against the direction of flow, and the trailing opening faces the direction of flow. The two openings are shown connected by a vertical U-tube containing mercury, or water, in Fig. 93.

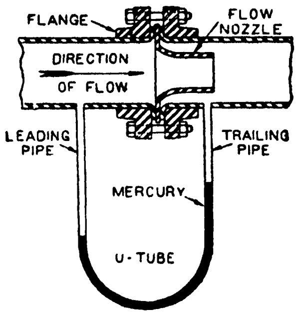


FIG. 93.—Principle of Pitot tube method.

The leading opening is subject to a static plus a dynamic or velocity head, the pressure there varying as the square of the velocity. The trailing opening is subject to a static pressure only. The difference in level of the liquid in the manometer limb is, therefore, a measure of the velocity head, and, therefore, of the velocity.

If h = this height or difference of level in feet of water, and v = mean velocity of the air in the main, in feet per second, then the following relation exists :—

$$v^2 = k \cdot h$$

where k is a constant, whose value for ideal steady flow across the section of the main is $2g$ (= 64·24).

In order to obtain the best average reading for normal velocities, a special Pitot tube of the type shown in Fig. 94 is employed. It is in the form of a nozzle plug which is inserted in the main and consists of a double conduit tube extending across the pipe diameter, each conduit having a different set of openings. The leading set of openings extends the whole length of the tube across the pipe diameter, and faces against the direction of flow. The trailing opening is located midway between the ends of the tube at the centre of the pipe diameter. This method enables the average velocity to be determined.

For high velocities a special design of nozzle plug is employed. In this nozzle plug the trailing opening is arranged to face in the same direction as the leading set of openings, namely, against the direction of flow, and a counter pressure is set up which reduces the total differential pressure by a definite amount. This reduction in the differential pressure enables the high velocity nozzle to measure about 50 per cent. more flow than in the preceding example.

For low velocity air-flow measurements it is often desirable to introduce a special smaller diameter pipe, which will give an increased velocity, with no appreciable drop of pressure. The velocity in the reducer is not increased above a value which is considered a good operating value for a pipe of the same diameter.

The nozzle-plug method of measuring the air flow is then applied to the reducer tube.

For pipes below about 2-inch diameter, Messrs. B.T.H. employ the orifice-tube method.

The orifice-tube consists of a pipe tapered internally at each end, so as to form a Venturi tube. The difference in head is measured between the smallest section (or throat) and the normal section represents the velocity head. Instead of employing the U-tube manometers in Fig. 93, it is now the practice to connect the two pressure tubes to a large dial instrument of the type as illustrated in Fig. 96.

The B.T.H. flowmeter illustrated in Fig. 96 is the indicating-recording type. The mechanism for this instrument is shown in Fig. 95, and consists of an iron casting for the body so designed as to form the limbs and well of a U-tube section similar to that shown in Fig. 93. A circular rack and pinion are employed to transmit the movement of the float. The rack engages a pinion mounted on a shaft carrying a horse-shoe magnet, which has its pole faces near and parallel to the inside surface of a copper plug fastened to the body of the meter. Another horse-shoe magnet is mounted on pivot bearings with its poles near and parallel to the copper plug, and with its axis of rotation in alignment with the shaft carrying the other horse-shoe. The indicating needle is attached directly to this outside magnet. A pinion on the shaft carrying the outside magnet engages the sector, the shaft of which carries the recording pen. The clock driving the recording chart is mounted in front of the magnet pinion and sector.

The circular chart on which the record is made is 12 inches diameter. A glass pen is employed which holds sufficient ink for one revolution of the chart, representing in the standard instrument 24 hours' running. The air-flowmeter for permanent installation is provided with a scale in cubic feet of free air (14·7 lb. sq. in. absolute) per minute at a temperature of 70° F. for a given pipe diameter, pressure, and temperature.

3. **The Electric Air-Flowmeter.**—This instrument was designed by Professor Callendar, F.R.S., for internal combustion engine use, and it has been used successfully upon aircraft engines in flight.

The principle of the method [1] depends upon the heating, by means of an electric current flowing through a resistance coil of the air in motion, and the measurement of the voltage, or potential difference, at the terminals of the coil. If V represents the applied voltage, and R the coil resistance in ohms, then for a weight W

[1] For a more complete account the reader is referred to the Aeronautical Research Engine Sub-Committee Report No. 55, September, 1920.

grammes of air flowing past the coil (and being heated by it) per second, we have the following relation :—

$$JSWdT = V^2/R,$$

where J is the mechanical equivalent of heat in joules per calorie, S the specific heat of the air, and dT the rise in temperature °C.

Then $$W = V^2/RSJdT.$$

The specific heat S does not vary appreciably (less than 0·5 per cent. for 100° C.), with temperature or pressure, JS is constant, and its value is very nearly unity.

Thence $$W = \frac{V^2}{RdT} \text{ very nearly.}$$

The resistance R is known at all temperatures (it can be arranged for this to be constant over the measured range of dT), so that it is only necessary to measure the voltage V necessary to produce a constant rise dT, in order to obtain the quantity of air flowing in unit time.

Thus $$W \propto V^2.$$

For practical measurements, it is found advantageous to employ a " resistance mat " type of coil in order to heat the air as uniformly as possible, and to measure the temperature rise by means of a pair of differential resistance thermometers of bare wire, the grids of which are uniformly wound and placed at equal distances on either side of the heating mat, so that any radiation errors are also balanced, and thus eliminated. This type of thermometer is very sensitive, and enables a temperature difference as small as 2° C. to be employed, thus reducing the expenditure of power required in the heating coil to about 2 watts per h.p. of the engine.

Gauze screens are placed across the tube on either side of the instrument in order to render the flow as uniform as possible. For the same reason the flowmeter should not be too large, but the air velocity should be as high as possible.

The difference of temperature between the two thermometers can be measured accurately by the Wheatstone bridge method. If the bridge is set initially at 2° C. out of balance, the heating current can be adjusted by means of a rheostat until the galvanometer needle returns to zero. The voltage required can then be measured. The most convenient instrument for this purpose has been found to be Professor Callendar's volt-thermometer, which employs a zero method of reading, and can be compensated for temperature effect with a high degree of accuracy. It gives also a practically constant sensitiveness at all air flows.

In this method the voltage is applied to the arms of a Wheatstone bridge in one branch of which is a very fine platinum wire. The change in resistance of this wire due to heating is a measure of the voltage applied.

Fig. 97 illustrates diagrammatically the arrangement of the air-flow meter, and Figs. 98 and 99 show the electrical connections.

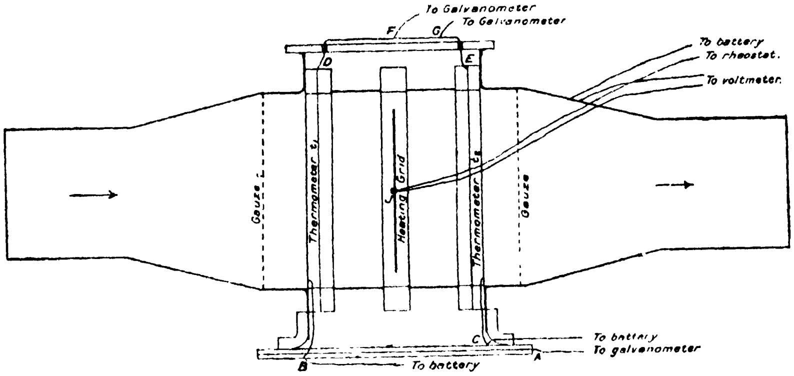


FIG. 97.—Showing principle of Callendar electric air-flow meter.

In the original form the body consisted of a 3-inch tube, 4 inches long; this was suitable for a 40 h.p. engine. At each end the diameter was reduced, as shown in Fig. 97, to 2 inches. The complete instrument was 12 inches long. Across the central section was the heating grid JK (made of an alloy known as "Climax") which had a resistance of 1 ohm.

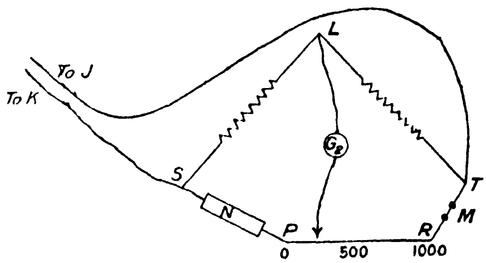


FIG. 98.

It was found that for air flows of 3 to 27 grammes per second a range of current of from 3 to 9 amperes was required; the resistance variation from the mean was only 1 part in 500. At $1\frac{1}{4}$ inches on each side of the heating grid the thermometers BD and EC were placed. In order to assist in the mixing of the air and to obtain a more perfect average temperature the wires of one grid were placed at right angles to those of the second grid of the same thermometer, and at 45° to the wires of the heating grid. The electrical resistance of

each thermometer was $43\frac{1}{2}$ ohms at 0° C. In the case of a flowmeter for 200 h.p., it is possible to obtain higher electrical resistance and sensitivity without using two grids for each thermometer. A piece of fine copper gauze was placed across the two ends of the body of the instrument to encourage uniform flow and to stop any fine particles.

The two grids AB and AC (of "Climax" metal) formed the equal arms of the Wheatstone bridge for measuring the temperature rise. The two other arms consisted of two thermometers which were connected together by a length of platinum wire DE, F being the balance point when no current was passing through the grid.

When the air current flowed, as shown in Fig. 99, and with the electric current passing through, the heating grid EC was hotter than BD and its resistance was higher. The new balance point is represented by H, the resistance of FH being one-half the increase of resistance of EC. Platinum connecting pieces were silver-soldered on to DE at F and H, so that each could, in turn, be connected through the galvanometer to A. The rise in temperature represented by FH was about $2\frac{1}{2}$°. The watts which could safely be applied to the heating grid were 100, so that, in this case, the maximum air flow measurable was 40 grammes per second (about the quantity required for a 40 h.p. engine). The flowmeter described is usually calibrated against an air-box orifice type.

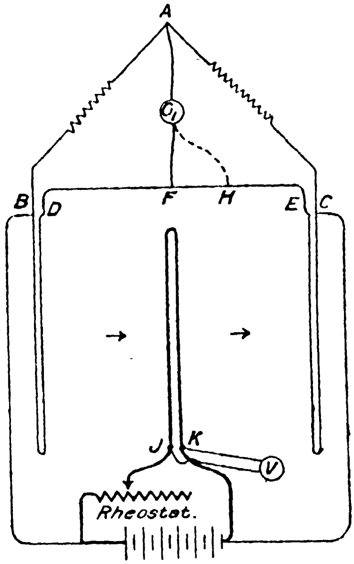


FIG. 99.

It has been successfully applied to the motor-car, air measurements having been obtained on a car engine on the track at Brooklands. The results showed that the readings of the air used by the engine obtained with the flowmeter agreed with those of the air-box to within 0·1 gramme per second, for flows up to 20 grammes per second. This represented the accuracy with which readings could be taken under road conditions on the car.

The flowmeter has also been applied to the measurement of the air supply of an aeroplane engine, in flight, at Farnborough, Hants. An instrument with a capacity range from 110 to 250 grammes per second was used in conjunction with a Siddeley Puma engine, in flights at altitudes up to 14,000 feet. The air consumption fell from 207 grammes per second at 1800 feet to 137 grammes at 14,000

feet, at corresponding engine revolutions of 1400 and 1320 r.p.m. respectively. A full account of these tests is given in the paper referred to in the footnote.[1]

Altitude Laboratory Methods.—In the tests carried out in the altitude laboratory of the American Bureau of Standards,[2] the amount of air flowing to the carburettor was measured in two ways, namely, with a Thomas electric flowmeter and with a Venturi tube. The former was specially built for the work, and consisted of a wooden box 6 inches square on the inside and 16 inches in length, containing a heating grid between two sets of thermocouples; this, it will be seen, is a somewhat similar method to the Callendar one previously described. The principle of operation was simply that a given energy input to the heating grid would cause a rise in temperature (measured by the thermocouples) inversely proportional to the mass flow of air.

The heating unit was merely a length of resistance wire strung back and forth across the middle of the box. In practice an E.M.F. of 60 volts was impressed on it, giving a current of 2·9 amperes. The thermal element consisted of 20 copper-constantan couples arranged in series, four junctions being encased in each of five stream-lined struts placed in each end of the box. The four couples were spaced equally down the length of the strut so that the result of all the couples gave an average for the temperature rise over the whole cross-section.

The Venturi meter mentioned was a large 6-inch one with a 3-inch throat; it was calibrated against a second Thomas electric flowmeter; the connections from the Venturi meter were carried to the manometer board outside the altitude chamber.

4. **Direct Air Measurement Method.**—Most of the air measurement devices described require initial calibration and occasional checking by some direct method of measurement; the calibrated gasometer method is probably the best for this purpose.

For original and research work it is an advantage to employ the direct method, using an air-flow meter as an indicator only. The method in question consists of a gas-holder of the inverted type, with a water seal. The internal volumes of the gas-holder corresponding to any height of the inverted bell above the water surface are obtained initially by direct measurement. When the engine or other object of supply is drawing its air from the gas-holder, the inverted bell falls progressively, and from readings of its positions at different intervals the corresponding volume of air used can be determined. It is necessary to know the pressure,

[1] "The Relationship of Air Consumption to B.H.P. in Internal Combustion Engines; Road and Flight Tests," Dr. H. Moss, *Proc. Inst. Mech. Engrs.*, April, 1924.
[2] *Vide* Chapter XII.

temperature, and humidity (or moisture content) of the air for this purpose.

The gas-holder employed for the air-box experiments mentioned in the lower footnote on page 130 is illustrated, externally, in the right-hand diagram of Fig. 100, whilst a part-sectional diagram is given on the left-hand side of the same illustration.

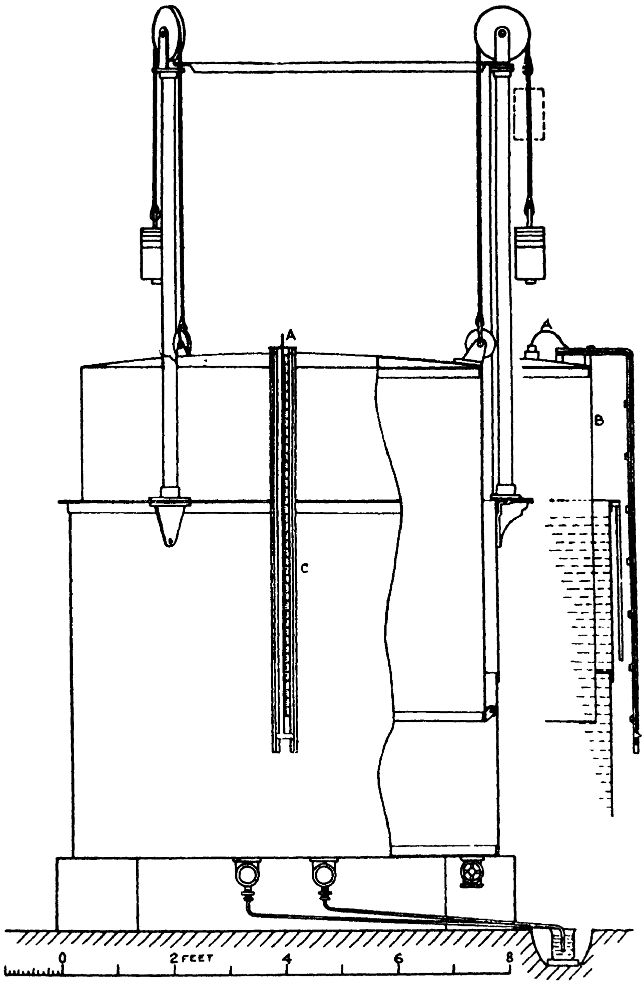


FIG. 100.—Gas-holder used for direct air measurements.

The water level inside was measured, in this case, by means of two water gauges, on opposite sides of the bell. The weight of the bell was balanced largely by means of three weights which were secured to it by flexible steel-wire ropes running over pulleys mounted

on ball-bearings. The guide pulleys running up the standards were also fitted in this way in order to reduce friction to a minimum. To promote circulation inside a small electrically-driven fan was fitted above the top of the bell, the motor-driven shaft working in an air-tight gland. A water-spray was also provided, by means of which the equalization of the temperature of the air was assisted, and at the same time complete saturation with water vapour secured. It was found that about 15 minutes were required for the incoming air to reach a steady state of temperature and humidity.

The temperature of the air inside the bell was given by two

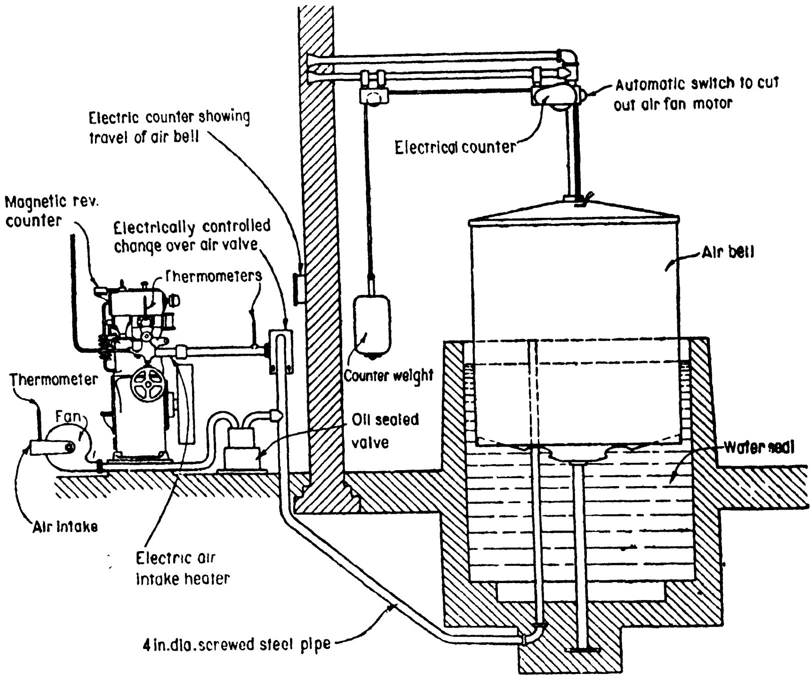


FIG. 101.—The Ricardo air-measurement apparatus.

thermometers (not shown in Fig. 100). The pressure difference between the interior and exterior of the bell was read off directly on a water-manometer. It is necessary when applying the direct method of air measurement to take into account the volume, not only of the bell, but also of the stand pipes and fittings; further, the volume occupied by the amount of the wall of the pipes projecting into the bell should be allowed for.

It will be apparent from what has been stated that simultaneous readings of the gas-holder position, internal pressure, and temperature are necessary during an air measurement test.

Ricardo has employed the gas-holder method for research and

other purposes.[1] Fig. 101 illustrates a layout of the apparatus employed.

The gas-holder consisted of a sheet-steel bell guided by a central column and balanced in the usual way. The lower end of the bell was partially closed, but three holes were left open for the water to pass through. These holes, which were of sufficient size not to interfere with the free flow of water under normal conditions, were effective in preventing surging.

The gas-holder was filled by means of the small, electrically-driven centrifugal blower shown on the left in Fig. 101. This fan delivered air through an oil seal (which acted as a non-return valve) into the holder, and continued to do so until the latter was full. The top of the bell then opened a switch automatically, and thus stopped the electric motor driving the blower. The stopping of the blower permitted the oil in the sealing box to flow back into

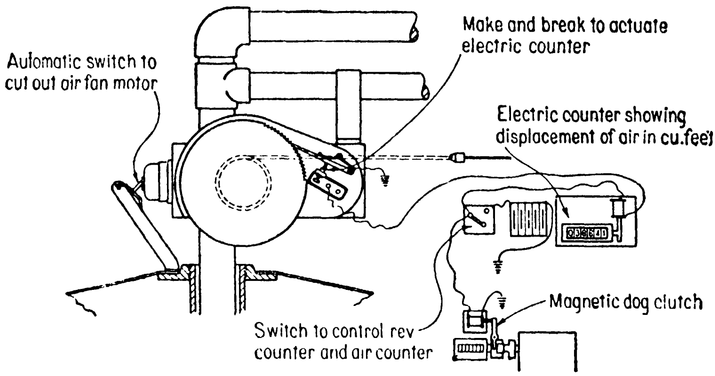


FIG. 102.—Electric counter used with Ricardo gas-holder apparatus.

the air inlet, and so prevented any air from being sucked through into the bell.

The amount of air consumed was recorded by means of a magnetically-operated counter, recording 0·2 cubic foot. Fig. 102 illustrates the general arrangement of this device.

A roller chain connected near the centre of the top of the bell passed over a sprocket pulley and was kept taut by the balance weight. A fine pitch ratchet wheel was mounted on the spindle of the chain sprocket wheel and driven by means of a light friction clutch so that it rotated with it as the bell fell, but remained stationary as it rose. The ratchet wheel operated a low-tension contact breaker in such a way that contact was made and broken by each tooth of the ratchet. This contact maker recorded on an

[1] "Some Recent Research Work on the Internal Combustion Engine," H. Ricardo, Soc. Autom. Engrs., U.S.A. Reprinted in *The Automobile Engineer*, September, 1922.

electric counter, placed on the operator's desk, the number of contacts made during the period the circuit was maintained. The ratchet wheel teeth were so designed that contact was made and the counter advanced one digit for every 0·20 cubic foot of air drawn from the holder.

A double-beat valve with rubber faces was provided for the purpose of connecting the engine with the atmosphere or with the gas-holder as required. It could be operated electromagnetically in a very quick manner, for the purpose of changing over the air supply.

The method of carrying out a fuel and air consumption test was as follows: When the engine was working normally, the fuel-measuring device [1] was filled, and the gas-holder charged with air. As soon as these were full, the observer threw over a switch and thus operated the double-beat valve, so that the engine drew its air from the holder and also its fuel from the measuring vessel. When the fuel passed the upper mark in the measuring vessel, the operator closed a switch and started a stop-watch. The former operation started the air counter device previously mentioned. At the end of the fuel test, i.e. when the fuel passed the lower mark on the measuring vessel, the switch was broken and the stop-watch stopped. The data obtained gave the total number of engine revolutions and the amounts of air and fuel in the stop-watch time period. From these values the mixture strength and fuel and air consumptions at the known engine speed were at once obtainable.

It is claimed that the variation in the readings obtained with different observers seldom exceeded 0·25 per cent. Errors due to changes of temperature of the air in the holder, due to weather effects (sunshine or rain, etc.), were reduced to a minimum by speed of operation. The holder was filled in about 30 seconds and its contents withdrawn in about 150 seconds, on the average, so that the total time from start to finish was about 3 minutes. Readings of the temperature of the air entering the bell, at the double-beat valve, and close to the carburettor, were taken; under ordinary atmospheric conditions the temperature seldom varied by more than 1·5° C. at these three points.

Fig. 103 illustrates diagrammatically a typical direct air-measuring arrangement,[2] in which a blower with a shut-off cock is employed to fill the gas-holder, the air supply to the engine passing through an expansion box and pipe. The gas-holder manometer shown enables the difference between the internal and external pressures to be measured, and, since the barometric height can be obtained, the true internal pressure can be found.

Alternative Methods for Measuring Volumetric Efficiency.—It is possible, in the case of single cylinder engines, to obtain a

[1] *Vide* Fig. 49.

[2] *The Automobile Engineer*, June, 1923.

fairly close approximation of the volumetric efficiency, and also air consumption, without actually measuring the air. The results of a very large number of tests made upon research engines, with different fuel-air ratios, have shown that at the mixture value for complete combustion the indicated mean pressure, using petrol, is 4 per cent. (with benzol, 5 per cent.) lower than the maximum value obtainable. The fuel consumption in both cases is 4 per cent. above the minimum value.

Fig. 2, on page 6, illustrates the results of numerous fuel consumption and indicated mean pressure tests obtained by Ricardo. On this diagram a vertical line has been drawn cutting the fuel consumption line 4 per cent. below the maximum i.m.e.p., to give the point of complete combustion mixture strength.

Knowing the weight of fuel consumed and the correct mixture

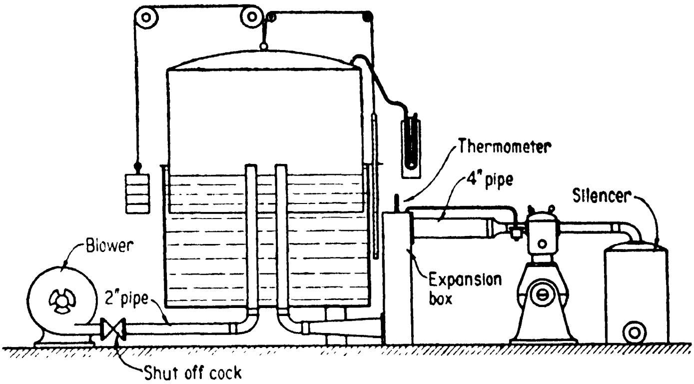


FIG. 103.—A direct air-measurement apparatus.

strength, the weight of air consumed and, therefore, the volumetric efficiency are determinable.

Again, the results of air measurements carefully carried out have shown that the weight of air consumed per I.H.P. per hour is virtually constant over a mixture ranging from 5 to 35 per cent. excess of fuel. If the carburettor is adjusted to give a rich mixture, and the I.H.P. is ascertained, it is possible to deduce the air consumption, and, from the known fuel consumption, the mixture strength also.

The Alcock Viscous Flow Air Meter.—A viscous flow air meter, designed by J. F. Alcock of the Ricardo Engineering Work, Shoreham,[1] possesses notable improvements upon the previous types. In particular it eliminates the serious errors which arise when pulsating air flows are measured in kinetic air meters of the orifice, Venturi and other types. These errors are partly accounted

[1] Described in *The Engineer*, December 30, 1938.

for by "root-mean-square" effects and partly by the flow in and out of the manometer, set up by pressure variations.

In the viscous flow air meter, the meter element is a honeycomb of long triangular passages, 3 inches long and 0·017 inch in height. Within the working range the flow through these passages is viscous, and the resistance of the element is therefore directly proportional to the velocity; this fact automatically eliminates the "root-mean-square" error. The "manometer connection" error is claimed to be eliminated by the special construction of the relevant parts, as described later.

The arrangement of the meter is as shown diagrammatically in Fig. 104. Air passes through two cleaner elements—one oily to trap dust and the other dry to trap any oil blown over from the first cleaner—and then through the meter element, which is made

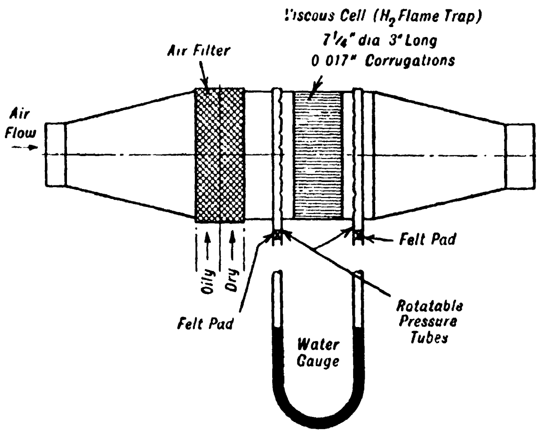


FIG. 104.—The Alcock viscous flow air meter.

up of alternate layers of flat and corrugated nickel silver strip metal wound upon a core. The manometer connections are tubes spanning the face of the element with a row of holes along one side, those on the upstream connection pointing downstream, and vice versa. This arrangement provides a reverse kinetic head which tends automatically to correct the small kinetic pressure drop in the element set up by entry effects. Felt pads in the connections render the flow therein viscous, thereby eliminating the "manometer connection" errors.

Experiments have shown that in a meter of this type, the manometer head is practically proportional to the flow velocity, and that even a very irregular flow, as when coupled to a single-cylinder engine without any smoothing, does not cause error.

Since the pressure difference is proportional to the flow rate the range of flow, measurable with reasonable accuracy, is much

greater than with kinetic types, where the pressure varies as the square of the flow rate. For example, suppose that at the maximum flow the manometer head is 30 cm. of water. At one-tenth of this flow the viscous meter gives a head of 3 cm. of water, which can be read with fair accuracy, whereas a Venturi meter would give only 0·3 cm. head of water, which is too small for accurate reading on a normal manometer.

Estimation of Air Quantities.—The installation of an air-measurement apparatus necessitates a knowledge of the amounts of air to be dealt with in order to assign the proper value to the capacity of the apparatus. The estimation of the air consumption is quite straightforward and involves a knowledge only of the engine speed and cylinder capacity.

Thus, if the total piston-swept volume be V cubic inches and the engine speed N r.p.m., then, since there is one suction stroke every two revolutions, we have

$$\left.\begin{array}{l}\text{Volume of air drawn}\\ \text{in at full throttle}\end{array}\right\} = e\,.\frac{NV}{3456}\ \text{cubic feet per minute,}$$

where e is the volumetric efficiency.

For most purposes the value of e may be taken as 75 per cent. at normal speeds, in the case of well-designed engines.

Example.—Estimate the weight of air at 20° C. temperature consumed by a four-cylinder petrol engine of 3·0-inch bore and 4·0-inch stroke, running at 1500 r.p.m. assuming $e = 0{\cdot}75$.

$$\left.\begin{array}{l}\text{The volumetric capacity}\\ \text{of the engine is given by}\end{array}\right\} \frac{4 \times \pi \times 9 \times 4}{4 \times 1728}\ \text{cubic feet,}$$

or

$$V = {\cdot}0655 \text{ cubic foot.}$$

$$\left.\begin{array}{l}\text{The volume of air}\\ \text{drawn in per minute}\end{array}\right\} = e\,.\frac{NV}{2}$$

$$= \frac{0{\cdot}75 \times 1500 \times {\cdot}0655}{2}\ \text{cubic feet}$$

$$= 36{\cdot}9 \text{ cubic feet.}$$

The volume per hour $= 36{\cdot}9 \times 60 = 2114$ cubic feet.

The weight of air per hour, taking the density of the air at 20° C. as 0·075 lb. per cubic foot is given by

$$W = 2114 \times {\cdot}075$$
$$= 158{\cdot}5 \text{ lb.}$$

For approximate purposes the following values of air densities and temperature can be used; their accuracy is sufficient for most practical purposes. For standard density values the table given in Appendix I should be consulted:—

TABLE X

Air Densities at Different Temperatures

Temperature in Degrees Fah.	Weight per cubic foot, in lb.	Volume per lb. in cubic feet
0	0·08633	11·583
32	0·08073	12·237
40	0·07944	12·586
50	0·07788	12·840
60	0·07609	13·141
70	0·07495	13·342
80	0·07356	13·593
90	0·07223	13·845
100	0·07094	14·096
150	0·06515	15·351
200	0·06021	16·606

Expressed in terms of the power output, the air consumption of a modern petrol type engine of compression ratio 5, running at full throttle on petrol, is approximately 6·00 lb. per I.H.P. hour; the mixture strength has little influence on this value over a 30 per cent. variation from the complete combustion value.

For ethyl alcohol, with the same compression ratio, the quantity of air per I.H.P. hour, when the engine is running on the complete combustion mixture, is approximately 5·8 lb.

The quantity of air used diminishes as the compression ratio is raised, as the following results, which are due to Ricardo, show :—

TABLE XI

Varying Compression Ratio with 20 % Rich Benzole Mixture

Compression Ratio	Lb. of Air per hour	I.M.E.P. lb.-sq. in.	I.H.P.	Lb. of Air per I.H.P. hour
4	200·5	125·0	30·3	6·62
5	194·0	136·5	33·1	5·32
6	188·0	145·0	35·2	5·34
7	164·0	152·0	36·8	5·00

Chapter V

WATER SUPPLY AND HEAT MEASUREMENTS

Quantity of Cooling Water.[1]—The following heat balance table represents, approximately, the manner in which the fuel energy is utilized in a petrol type engine :—

Heat available as B.H.P. .	24
,, absorbed by engine friction	4
,, carried off by exhaust gases	40
,, ,, ,, cooling water	28
,, lost by radiation	4
	100

These values will, of course, vary with the type and thermal efficiency of the engine ; and with the nature of the fuel. They will also depend to some extent upon the throttle opening. If C denote the calorific value of the fuel in B.T.Us. per lb. and w the petrol consumption per B.H.P. hour, in lb.,

$$\left.\begin{array}{r}\text{Then total heat equivalent}\\ \text{of fuel used per hour}\end{array}\right\} = w \times \text{B.H.P.} \times \text{C.} \quad \text{B.T.Us./hr.}$$

From the above table it will be seen, therefore, that

$$\left.\begin{array}{r}\text{Heat carried off by}\\ \text{cooling water } H_w\end{array}\right\} = 0{\cdot}28\, w.\text{C. (B.H.P.).} \quad \text{B.T.Us./hr.}$$

Taking the calorific value of petrol at 18,600 (B.T.Us./lb.), we have

$$H_w = 5208\, w \times \text{B.H.P.} \quad \text{B.T.Us./hr.}$$

Example.—It is required to know what quantity of cooling water is necessary for a 40 h.p. petrol engine, the inlet and outlet temperatures of the water being 60° F. and 180° F. respectively, and the fuel consumption 0·50 lb. per B.H.P. hour.

$$\begin{aligned}\text{Then} \qquad H_w &= 5208 \times 0{\cdot}5 \times 40\\ &= 104160 \quad \text{B.T.Us./hr.}\end{aligned}$$

Now, if Q be the quantity of water used in lb. per hour, and T_i and T_o the inlet and outlet temperatures in ° F.,

$$\begin{aligned}\text{Then} \qquad Q(T_o - T_i) &= H_w.\\ \text{Thence} \qquad Q(180 - 60) &= 104160.\\ Q &= \frac{104160}{120}\\ &= 868 \text{ lb., i.e. } 86{\cdot}8 \text{ gallons per hour.}\end{aligned}$$

[1] For automobiles it is usual to allow 1 gallon of water for every 5 to 7 B.H.P. with pump circulation, and from 30 to 50 per cent. more for thermosyphon cooling.

Another way of expressing the quantity of cooling water heat for approximate purposes is as follows :—

$$H_w = 2600 \text{ B.T.Us. per B.H.P. per hour.}$$

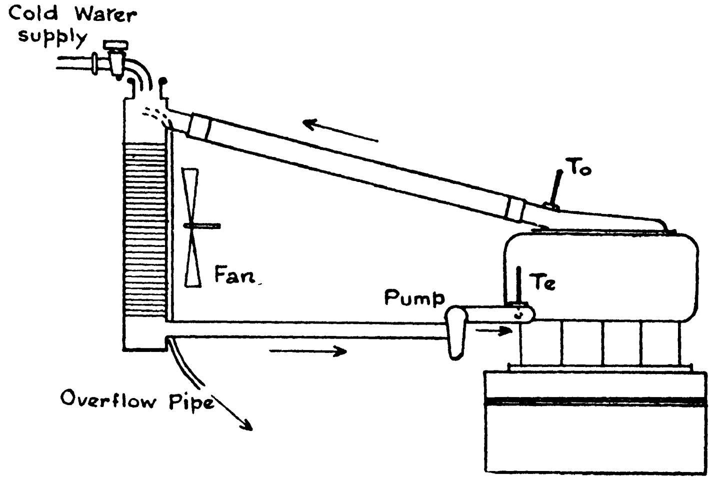


FIG. 105.—A simple water-cooling arrangement.

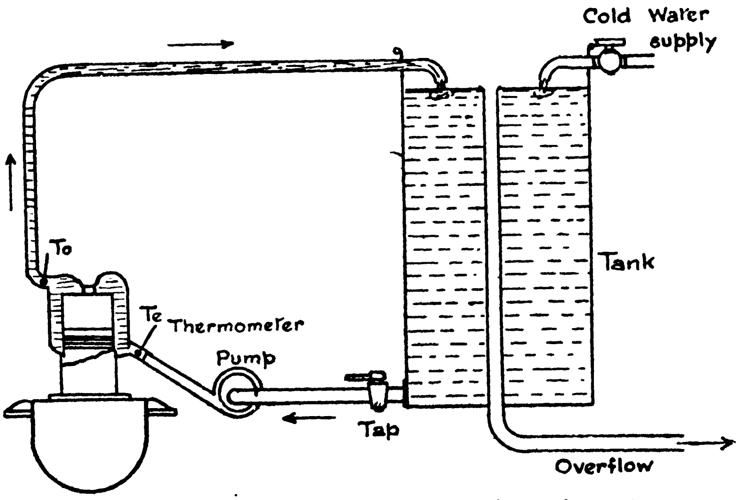


FIG. 106.—Cooling water arrangement for engine tests.

In the case of engine tests carried out in the laboratory, means must be provided to eliminate from the cooling system this quantity of heat. The arrangements shown in Figs. 105 and 106 enable the surplus heat to be got rid of effectively, so that the mean working temperature of the engine is maintained at a constant value.

Cooling Water Arrangements.—In petrol-type engine tests it is necessary to arrange for an adequate supply of cooling water to the cylinder jackets, and to provide means for measuring both the temperatures and quantity of the water used in a given time. It is also advisable to have a convenient means of regulating the supply and its temperature.

One of the simplest arrangements of the cooling water system for test purposes is that illustrated in Fig. 105, and in which the water is merely circulated around by means of a centrifugal pump, the speed of which can be varied. The capacity of the radiator is such that, together with the pump regulation, the desired range of water temperatures can be obtained. An air fan is used to assist the cooling of the radiator, and its speed can be varied if required.

In order to measure the quantity of water used in a given time, a water meter, or water-flow meter, can be employed. It is not necessary, here, to describe typical water meters, but it may be mentioned that these resemble the liquid flow meters already described.

Fig. 106 illustrates a satisfactory arrangement of the water system employed by the author in connection with research work upon an aircraft engine. A large galvanized tank acted as the main reservoir, and was provided with a cold-water supply, an overflow (to keep the head constant) and a hot-water discharge from the engine cylinder jackets. A centrifugal pump maintained the circulation in the direction shown by the arrows. Thermometers T_e and T_o were arranged at the inlet and outlets to the jackets.

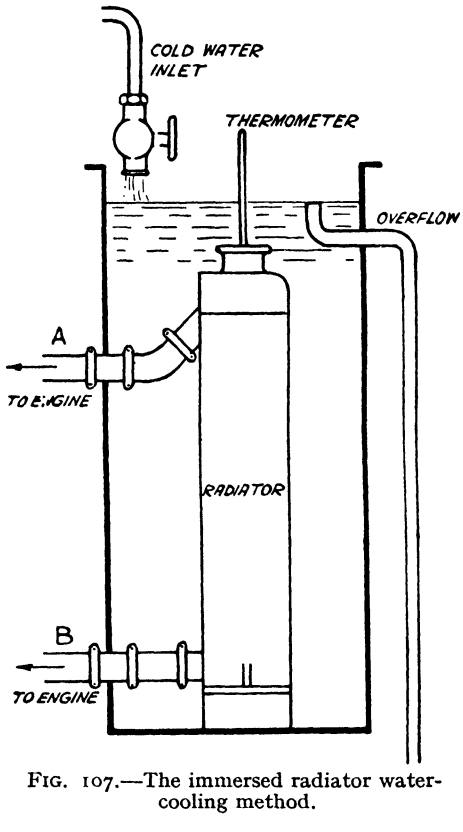


FIG. 107.—The immersed radiator water-cooling method.

Immersed Radiator Method.—Although it is frequently possible to employ an ordinary motor-car radiator, with fan-supplied cooling air, for engine tests in the shop or laboratory, it is not always possible to cool the water adequately; moreover, the use of cooling fans is not always convenient.

An alternative method and one that has been employed by one of the leading commercial motor firms is to immerse the radiator in another tank of water and to maintain a steady stream of cold water from the mains water system into this tank; an overflow is provided for the purpose of keeping the level constant.

A thermometer can be inserted in the top of the radiator, the cap of which is a water-tight fit, or better still, two thermometers, one at A and the other at B (Fig. 107), can be employed to give the outlet and inlet water temperatures of the jacket water.

It is a further advantage to fit a thermostat of the by-pass type in the upper water connection from the engine to the radiator so as to maintain the engine jacket cooling water at as uniform a temperature as possible.

Quantity of Water.—Referring to Fig. 106 the quantity of water at any given pump speed could be measured by directing the discharge into a previously calibrated vessel, and noting the time.

The amount of heat H_w extracted by the cooling water in given time t seconds is given by the relation

$$H_w = Q'(T_o - T_e) \, . \, t \quad \text{very nearly}$$

where Q' = lb. of water flowing through engine per second, T_o and T_e being the outlet and inlet temperatures respectively.

In measuring the quantity Q', the temperature of measurement should be noted, and a correction applied, if the volume method is used.

Thus, if V_t is the measured volume at temperature t_1°C., then volume at mean temperature $\frac{T_o + T_e}{2}$ is given by

$$V_m = V_t \, . \, \frac{1 + \alpha \, . \left(\frac{T_o + T_e}{2}\right)}{1 + \alpha t_1}$$

where α is the mean coefficient of expansion between these two temperatures.[1] Another method of measuring the quantity of water supplied is illustrated in Fig. 108. It is very similar to the arrangement adopted by The Institution of Civil Engineers in connection with their classical tests on gas engines in 1904-6.

In this case the water is supplied from a tank above the engine provided with a ball float valve for keeping a constant level. The water flows under this head to the jacket, an inlet thermometer being provided near the cylinder. The heated water flows out into a calibrated vessel upon a weighing machine. The temperatures of the outlet water, and of the water in the weighing machine vessel are measured.

[1] Values of the densities of water at different temperatures will be found in Kaye and Laby's Tables (Longmans, Green & Co.).

It is possible in this manner to measure the total amount of heat lost to the cooling water, including the radiation loss from the engine surface.

Radiation Loss.—The measurement of the radiation loss from an engine is a difficult one for the engineer to make, and the results obtained are usually approximate only.

The radiation loss in the case of a petrol-type engine is of the order of 4 to 6 per cent. of the total heat of combustion of the fuel.

An approximate method of determining the radiation loss [1] is to run the engine light, and then to adjust the number of explosions per minute so that the temperatures maintained on the thermometers measuring the temperatures of the water to and from the jackets are respectively equal to those obtained at a full-load

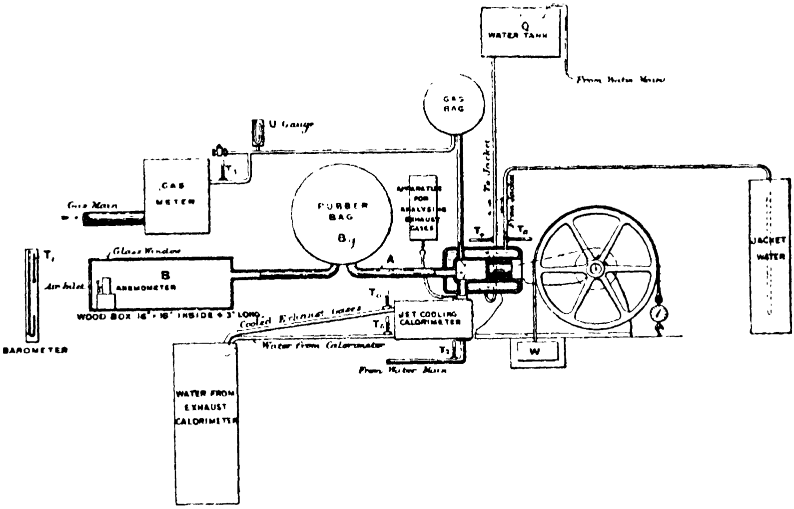


FIG. 108.—Apparatus used to measure jacket water quantity and heat content in I.C.E. tests.

trial. In this way the surface temperatures, allowing a very small quantity only of water to flow through the jackets in order to obtain similar conditions to those during the normal working of the engine, are kept approximately the same in the two trials, although the inner temperatures will be lower, due to the fewer explosions. Hopkinson determined the radiation loss, approximately, by determining the heat taken away by the jacket water when the exit temperature was a certain value, say 70° C., as compared with the engine running under exactly the same full-load conditions with the exit temperature at another value, say, 40° C. In the case

[1] The radiation loss in this case includes the radiation from the hot surfaces of the cylinder, piston, etc., and also that from the bearings of the engine due to friction, i.e. the direct radiation loss plus the difference (expressed in heat units) between the I.H.P. and B.H.P.

cited, he found the difference to be between 2 and 3 per cent. of the total heat supply; that is to say, between 2 and 3 per cent. of the total heat supply less finds its way with the jacket hot and jacket cold.

This premises, of course, that the heat flow from the hot gases to the walls does not vary with the exit temperature of the water jacket—which is not wholly correct.

Ricardo's Water Circulation Arrangement.—Ricardo has employed the arrangement illustrated in Fig. 109 for general engine tests.

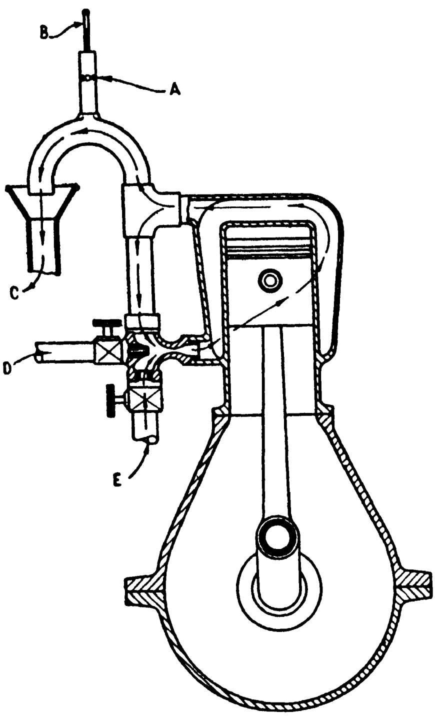


FIG. 109.—Ricardo water circulation method.

It consists of an injector device, comprising a small high-pressure jet of water which, by its kinetic energy, sets up a rapid circulation in a closed circuit as shown by the arrows. The Venturi high-pressure supply is at D, whilst the low-pressure supply is shown at E. There is an overflow pipe at C, the highest point in the system, an air vent A, and a thermometer B.

This arrangement provides a brisk circulation, and hence maintains a fairly uniform temperature. Moreover, the temperature of the cooling water can be controlled very effectively by the amount of cold water admitted through the supplementary supply E.

The total quantity of water in the system is only that required to fill the cylinder jackets and a little additional piping, so that its heat capacity is very small. Any desired temperature can, therefore, be obtained within a few minutes after starting, and the temperature can be varied rapidly and with precision.

The arrangement described, does not, as it stands, however, enable heat quantities to be measured, but is principally of use in providing an effective cooling water regulator, the temperature of which can be varied quickly. It enables also the best working temperatures to be attained and measured.

The Heenan Water Cooler.—The subject of water cooling has been studied carefully by the manufacturers of the Froude hydraulic

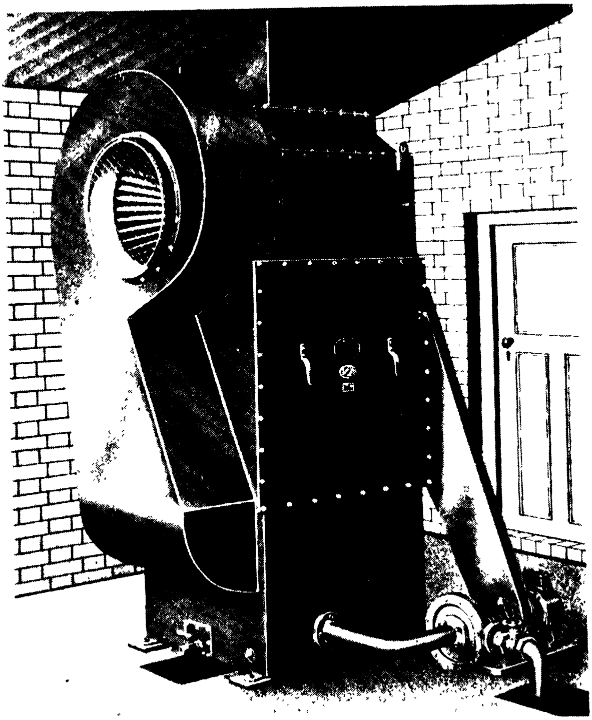
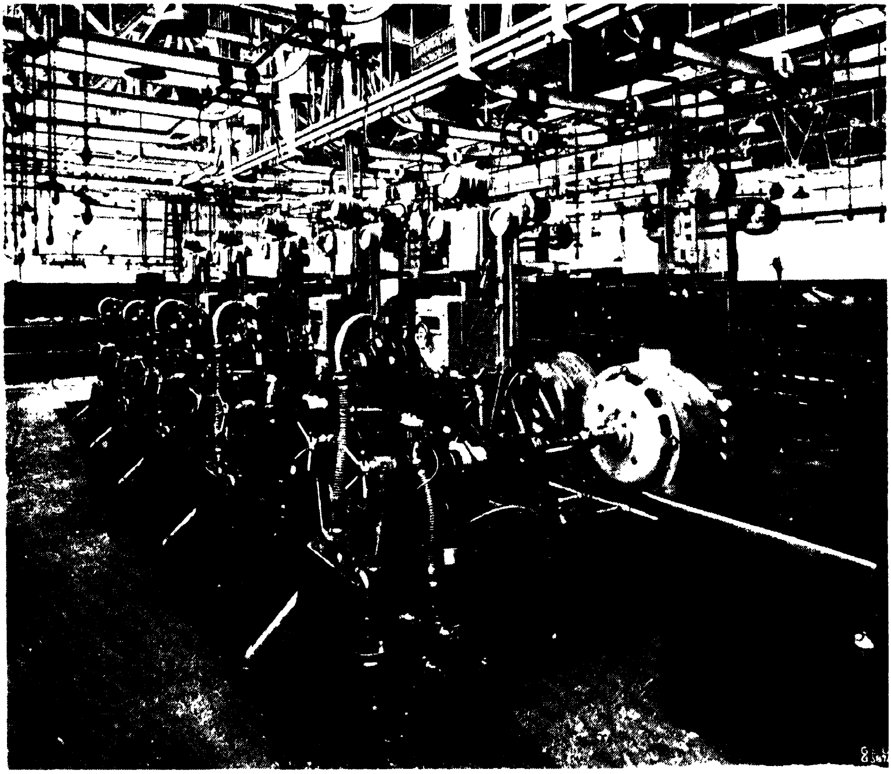


FIG. 110. The Heenan water cooler.

[See page 155.



[Courtesy Messrs. G. L. C., Ltd.

FIG. 111.—Hillman Minx engine test-bay, showing also water cooling arrangements.

[To face page 155.

absorption type dynamometers, for not only has the cylinder jacket water to be cooled, but the water passing through the dynamometer, or water brake, has to be kept below a certain temperature, since it heats up when the brake is in use.

A system has been evolved whereby the engine and brake-water systems are continuously re-cooled and re-circulated by means of the Heenan water cooler shown in Fig. 110. The cooler consists of a rectangular casing built up of heavy mild steel plates, and containing six tiers of cooling screens, having a cooling surface of galvanized mild steel. The hot water, after passing through strainers contained in the overhead tank, is evenly distributed over the cooling surface; it then falls into the tank formed by the lower part of the casing. Air under pressure is blown from the external atmosphere into the space beneath the cooling screens, through which it rises, cooling the water and afterwards passing to atmosphere by way of the vertical outlet duct at the top of the cooler.

The cooled water is delivered by a centrifugal pump either to an overhead storage tank, or direct to the pipes feeding the engines and hydraulic dynamometers; after becoming heated therein, it returns through the visible circuit to the inlet tank of the cooler.

The system described is very efficient, as it not only provides a definite uniform cooling effect, but occupies a comparatively small space compared with other water-cooling systems; moreover, it saves about 95 per cent. of the circulating water.

Engines which are cooled by water at a pressure above atmosphere or by a glycol-water mixture require a special layout of cooling plant. The "coolant" is passed through a heat exchanger which abstracts heat from it and transfers this heat to the cooling water, which is in turn cooled in its passage through the Heenan cooler. The latter also receives the heated water direct from the hydraulic dynamometer.

Measurement of Exhaust Heat.—Except in the case of special research work, it is seldom necessary to measure the actual amount of heat carried off, in a given time, by the exhaust gases. It may be of interest, however, to indicate the principles of the method which has been employed satisfactorily for this purpose. One cannot do better, in this case, than to refer to the method employed by The Institution of Civil Engineers in their classical tests.[1] The arrangement adopted is illustrated in Figs. 108, 112 and 113. The exhaust gases were led into an exhaust-gas calorimeter, and were cooled in passing through this apparatus. The calorimeter was water-jacketed right up to the engine exhaust flange, and the water (of the calorimeter jacket) after circulating through the jacket, was led to a rose form of spray from E, Fig. 113, through the

[1] "On the Limits of Thermal Efficiency in Internal Combustion Engines," *Proc. I.C.E.*, vol. clxix, p. 157 *et seq.*

small orifices of which it spurted out to meet the stream of exhaust gas. The water and cooled gases then found their way out, passing various obstructions to ensure mixing and abstraction of the heat from the gas until finally the gas, cooled down to about 90° F., escaped to the atmosphere, the water also escaping and having about the same temperature as the escaping gas.

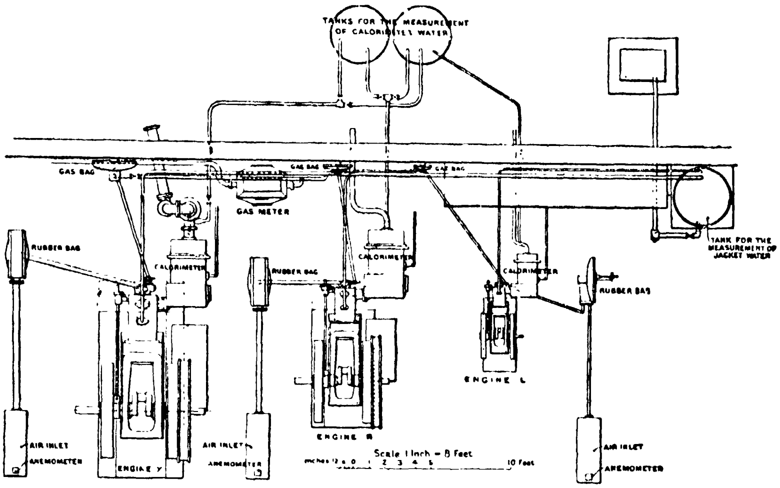
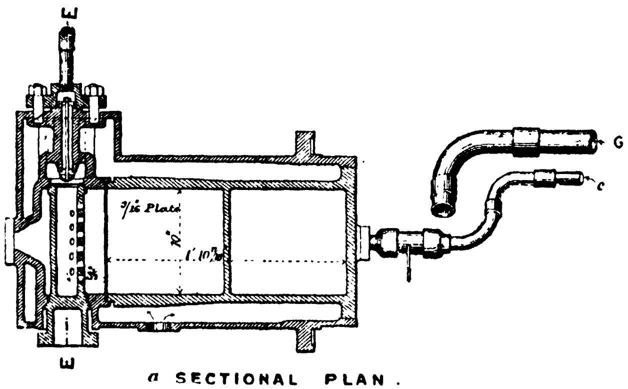


FIG. 112.—I.C.E. test apparatus.



a SECTIONAL PLAN.

FIG. 113.—Exhaust gas calorimeter.

The general arrangement of the calorimeter is shown in Fig. 112, from which it will be seen that the water was brought to the calorimeter directly from the water main, and was led to measuring tanks placed outside the test house. These tanks were fitted with gauges and scales graduated in feet and decimals of a foot, and were previously calibrated by pouring in weighed quantities of water.

In designing this type of calorimeter, care should be taken to ensure that the thermometer measuring the temperature of the escaping water is placed so that its bulb is completely immersed in the water. As the pipe does not always run full, a pocket should be formed in the pipe to ensure this condition.

The heat extracted by the calorimeter from the exhaust gases is obtainable from the measurements of the quantity of water flowing through the calorimeter and the rise in temperature of the water.

Thus, if Q lb. of water be used per hour, and if the inlet and outlet water temperatures be T_i and T_o° F. respectively, then

$$\left.\begin{array}{r}\text{Heat carried off}\\ \text{by exhaust gases}\end{array}\right\} = Q(T_o - T_i) \text{ B.T.Us./hr.}$$

If the exhaust gases are not at the temperature of the atmosphere on escaping from the calorimeter a correction must be applied; other corrections are necessary for radiation losses, the moisture present in the exhaust gases, due to combustion, etc.

Sir Dugald Clerk [1] has devised an improved type of exhaust calorimeter, in which the outlet thermometer is always completely immersed in the outlet water, and the gases are prevented from carrying away water mechanically suspended.

[1] "The Gas, Petrol, and Oil Engine," Dugald Clark, vol. i, p. 261.

Chapter VI

MEASUREMENT OF BRAKE HORSE-POWER

As the measurement of power output is one of the most frequent and important tests carried out upon high speed internal combustion engines, it is proposed to consider in some detail the usual methods employed. The measurement of power involves that of the work done in a given time, and since the work done represents the product of force by distance, or torque by angular travel or movement, it is necessary to measure three factors, namely, time, force, and distance, or time, torque, and angle. The apparatus employed for measuring power is termed a dynamometer, or brake, and it may be mentioned that there is now a number of such dynamometers available for various purposes.

Of these dynamometers, however, a limited number only is suitable for the testing of engines of the type here considered. Before proceeding to detailed description of some of these, it may be as well to mention the principles of the methods of measurement.

If a force F lb. acts through a distance d feet in time t seconds, then

$$\text{Horse-power exerted} = \frac{\text{F} \cdot d}{550t}.$$

Thus, it is necessary to know F, d, and t in order to obtain the horse-power.

In many cases, as with rope and Prony brakes, the distance d represents the peripheral distance of the force F acting on a flywheel or drum, and the expression $\frac{d}{t}$ is the peripheral velocity v.

In the case of a drum of diameter D feet, rotating at N r.p.m., we have

$$v = \frac{\pi \text{DN}}{60} \text{ f.s.},$$

so that the horse-power (h.p.) $= \dfrac{\pi\text{DNF}}{33000} = k \cdot \text{N} \cdot \text{F}$

where k is a constant.

It follows that a measurement of the force F (exerted at the periphery of the flywheel or drum) and the speed N of the latter is all that is required for power determination.

If the torque T in lb.-ft. of a power unit is measured, and if

θ is the angular distance in radians travelled in time t seconds, then the horse-power given out is

$$\text{H.P.} = \frac{\text{T} \cdot \theta}{550t}.$$

In the case of torque-reaction dynamometers, T is directly measurable, and equals the product of a weight F lb. acting at a known distance or radius d ft. $\left(= \frac{\text{D}}{2}\right)$.

Thus $$\text{T} = \text{F} \cdot d \text{ lb.-ft.}$$

Further, the angular velocity $w = \frac{\theta}{t} = \frac{2\pi\text{N}}{60}$ radians per second.

Hence $$\text{H.P.} = \frac{\text{T} \cdot \theta}{550t}$$
$$= \frac{\text{T} \cdot w}{550} = \frac{2\pi\text{N} \cdot \text{F} \cdot d}{33000} = \frac{\pi\text{DNF}}{33000},$$

which is the same expression as before.

Types of Dynamometer.—The dynamometers suitable for high speed internal combustion engines may be divided into two principal classes, as follows :—

(*a*) *Absorption Dynamometers,* in which all of the power output of the engine is absorbed by the dynamometer, and

(*b*) *Transmission Dynamometers,* in which the power is not absorbed, but merely transmitted without appreciable loss by the dynamometer, and is available for any useful purpose. The engine output can thus be measured whilst the engine is fulfilling its ordinary duties.

Absorption dynamometers are probably the cheapest and most convenient for most purposes, but the latter type enable power measurements, or records, to be made without interfering with the output. In this way it is possible on aircraft or automobile engines, etc., to carry out power tests under the actual working conditions for which these engines are designed. Transmission dynamometers are useful, also, in the case of engine tests of long duration, for economical reasons.

Absorption Dynamometers.—These may be divided into four principal classes as follows :—

1. Friction dynamometers, and of which the Rope, Prony, Flexible Band, and Clutch types are typical examples.

2. Hydraulic or water dynamometers, in which the power is absorbed by the churning, or eddying, and heating of water. The Froude, Brotherhood, and Junker's absorption brakes come under this category.

3. Air dynamometers, or brakes. In this type the power is absorbed in the churning, or eddying, and heating of the air. Typical examples are the Walker Fan Brake and the Aerial Propeller, or Airscrew types.

4. The Magnetic, or Eddy Current Brake. This depends upon the eddy currents set up by a special rotor, with copper portions, revolving in a strong, magnetic field. It is not used to any appreciable extent for the larger powers.

The particular type of dynamometer applicable, in any case, will depend largely upon the nature of the tests to be carried out, the size and type of engine, and similar circumstances. The detailed descriptions of the more representative types will no doubt enable the most suitable type to be selected. It is proposed to describe a few typical dynamometers following the above order.

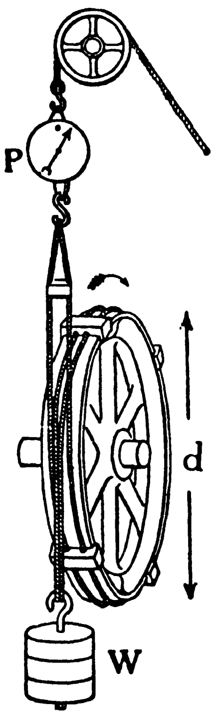


FIG. 114.—The rope brake.

Friction Dynamometers.—*The Rope Brake.*—This type comprises a simple form which can be used directly upon the periphery of a drum or flywheel attached to the engine shaft. It is an inexpensive dynamometer arrangement which is occasionally employed for rough determinations of power from small engines; its advantage for this purpose lies in the fact that it can readily be constructed in any workshop, without the need of any special appliances.

The power of the engine is absorbed as frictional heat created between the rope and the pulley surface.

If the pull in the spring balance rope be P lb. (Fig. 114), and the distance of the centre of the rope from the centre of pulley be $\frac{d}{2}$ feet, the weight on the lower end of rope being W lb. and the pulley speed N r.p.m., then the horse-power at the pulley is given by

$$\text{H.P.} = \frac{\pi(W - P) \cdot d \cdot N}{33000} \qquad \pi = 3{\cdot}14159.$$

It is usual to adjust the friction between the rope and pulley so that P is very small compared with W; any inaccuracy in measuring P is, therefore, unimportant.

The principal disadvantage of this type is that it is rather erratic in action, due to changes in the frictional coefficient, and that the flywheel or pulley rim becomes unduly heated in many cases.

Dynamometers of the cooled-rim type are occasionally employed for motor and engine tests (Fig. 115).

The Prony Brake.—In its simplest form this consists of a brake acting on a pulley, or flywheel, of the power unit under test. The

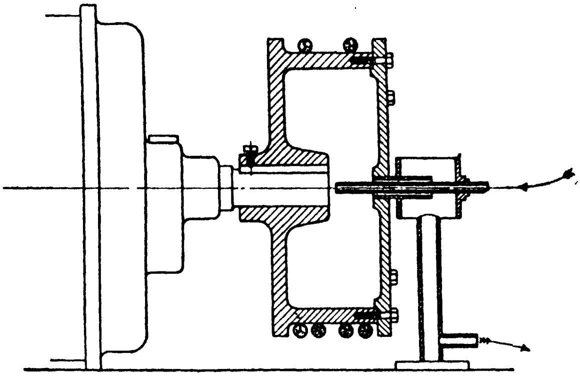


FIG. 115.—Water-cooled rope-brake drum.

force W lb. necessary to restrain the brake from being rotated by the pulley is measured, and knowing its leverage (or radius *d* ft.).

The horse-power is given by the relation

$$\text{H.P.} = \frac{2\pi N \cdot Wd}{33000}.$$

It is necessary to provide means for adjusting the friction of the wooden blocks, shown in Fig. 116; this is accomplished by means of the bolts and nuts shown.

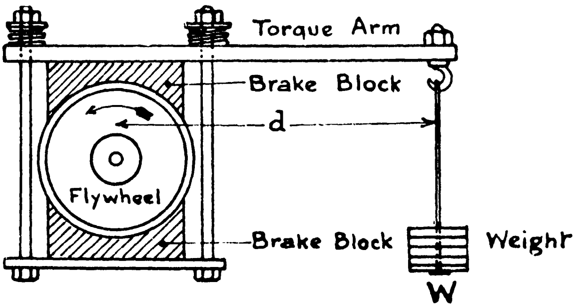


FIG. 116.—The Prony brake.

With a weight of 100 lb. and a lever arm of 6 feet it is possible to absorb 114·3 h.p. at 1000 r.p.m.

This type has been employed for measuring horse-powers up to about 200, and with special means for cooling the flywheel rim, was used by Thurston to absorb 540 h.p.

In its improved form the upper block is replaced by a flexible

band having a number of smaller blocks, together with means for damping out, or preventing any oscillation, or "snatching." Dr. Coales has designed the improved self-adjusting type of Prony brake, shown in Fig. 117.

This design obviates the unsteadiness in the spring balance readings of the usual type, due to variations of friction between the brake and brake-pulley. More especially is this unsteadiness present in the case of small single cylinder petrol engines, in which there are appreciable variations in the crank-effort.

The type of Prony brake illustrated in Fig. 117 obviates the use of a spring balance by a simple device which compensates the variations in friction between the brake and brake pulley and adjusts automatically the pressure of the brake belt, so that the

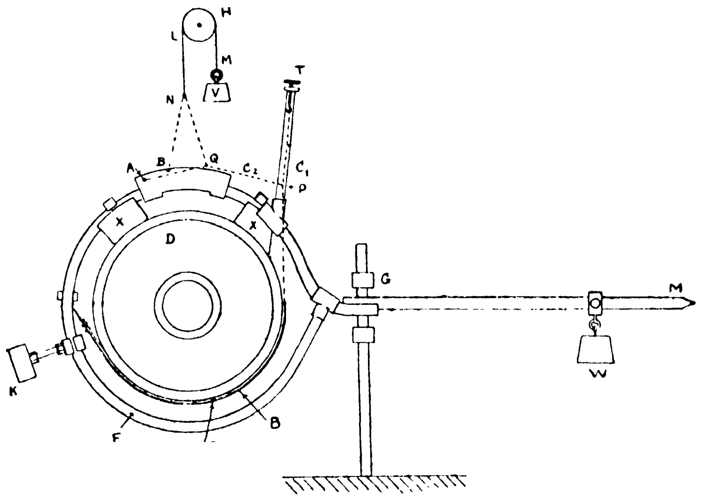


Fig. 117.—A self-adjusting Prony brake.

torque is maintained constant in value independently of the speed over wide limits. The torque is applied and measured by dead weights, no spring balance being used.

Referring to Fig. 117, D is a water-cooled pulley, in which the heat caused by the friction is dissipated by the boiling away of the water, which is replenished periodically. F is a tubular frame which supports the brake blocks XX, the flexible brass belt B, and the flexible cotton belt S, which lies between the belt B and the face of the pulley. K is a counterpoise to bring the C.G. of the apparatus, exclusive of the weight W, to the axis of the pulley.

The torque is varied by moving W along the graduated arm. The flexible brass belt B is attached by a chain C to a tension screw T by means of which the pressure of the belt S is adjusted so as to give the frictional drag on the pulley D.

A second chain C_2 is attached to the point P in the chain C_1, its other end being fixed permanently to a point A on the brake frame F. To a second point B on the frame and to the point Q on the chain C_2 a lighter chain BNQ is attached, and to a link on this a cord NM is hooked and carried over a pulley H and supports a weight V. The pulley H is supported on an adjustable arm attached to the stop G.

If the brake rises owing to an increase of friction, the tension due to the weight V is partly taken off NQ, and thrown more on NB; the tension of the chain C_2 is thereby reduced, and, therefore, also the component of the force which it exerts at right angles to the chain C_1. Thus the tension of C_1 is diminished to a still greater degree, and with it the pressure exerted by the brake belt on the pulley D. The brake torque is thus restored to its original value. Similarly, if the brake friction decreases, the reverse action occurs.

When the load is set by adjusting the position of the weight W, the tension of the chain C_1 is varied by the tension screw T until the brake arm is horizontal. The torque exerted by the brake is then equal to the product of the weight W and its distance from the centre of the pulley; this distance is read off the graduated arm. The pull exerted by the weight V in the cord LN does not exert any torque on the brake because the link at N into which the cord LN is hooked is such as to make the direction of LN pass through the axis of the pulley. Further, since V is only $\frac{1}{8}$ to $\frac{1}{10}$ W, even if it is not quite in line with the centre, its effect is less than 1 per cent. of the torque due to W.

This brake is particularly applicable to small internal combustion engines, on account of its automatic torque adjustment. It is made[1] in standard forms for h.ps. up to 5 and 12 respectively, and is quite inexpensive. It can be supplied also in larger sizes to order.

The Torque Reaction Method.—The majority of dynamometers now employ the method of torque reaction for measurements of the power absorbed, or transmitted.

The simplest method of employing this principle consists in making the frame or casing of the power absorption part of the apparatus capable of rotating about the axis of the dynamometer. This movement is restrained by hanging weights on to the end of a lever attached to the casing.

Alternatively, a spring balance, suitably calibrated for the purpose, can be used to counteract the turning moment and to measure the torque-arm load.

Where large forces have to be measured in this manner it is usual to employ some form of weighbridge to enable accurate measurements of these forces to be made. Fig. 118 illustrates the principle of the torque meter used on the S and F type Froude hydraulic

[1] Thos. Wyatt, 279 Deansgate, Manchester.

dynamometers. The casing, supported on anti-friction trunnions, not shown in the diagram, is provided with two lugs at diametrically opposite points on the periphery; each lug is coupled by a connecting rod, having articulated joints, to a system of levers fixed in the base-plate. When the direction of rotation is counter-clockwise as shown, the left-hand connecting rod presses downwardly upon the short arm of a main lever, pivoted upon a fulcrum which is connected by a rocking-pin joint to the bedplate. The right-hand connecting rod pulls vertically upwards upon the short arm of a similar main lever, which is pivoted upon a fulcrum permanently

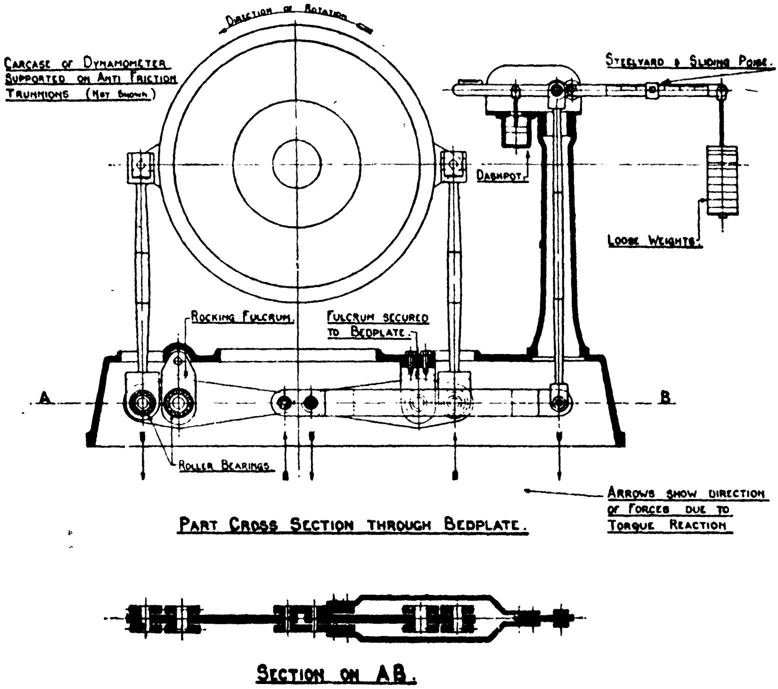


FIG. 118.—The Heenan and Froude torque-arm apparatus for dynamometers.

fixed to the bedplate. The long arms of the main levers, therefore, exert respectively upward and downward forces, and their ends are connected together by a third lever transmitting the resultant of these two forces to the short arm of a steelyard type of weighing machine. This system of levers is equivalent in effect to a lever arm directly attached to the dynamometer casing, and having an effective length several hundred times as great as the radius from centre of shaft to centre of one of the casing lugs. The torque can therefore be measured by a number of very small weights and a sliding poise instead of enormously heavier weights.

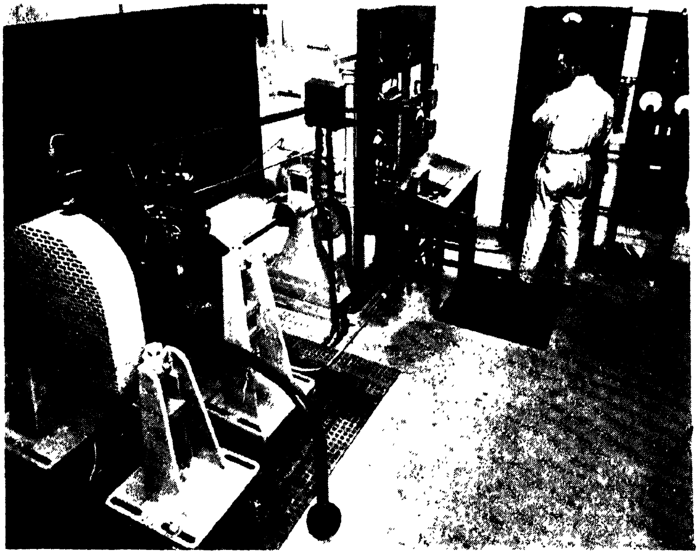


FIG. 119.—One of a series of dynamometer rooms for research purposes at the Vauxhall Motor Company's works. The panels carry electrical controls for varying load and speed, flowmeters, thermometers, oil and water temperature regulators, etc., conveniently grouped together.

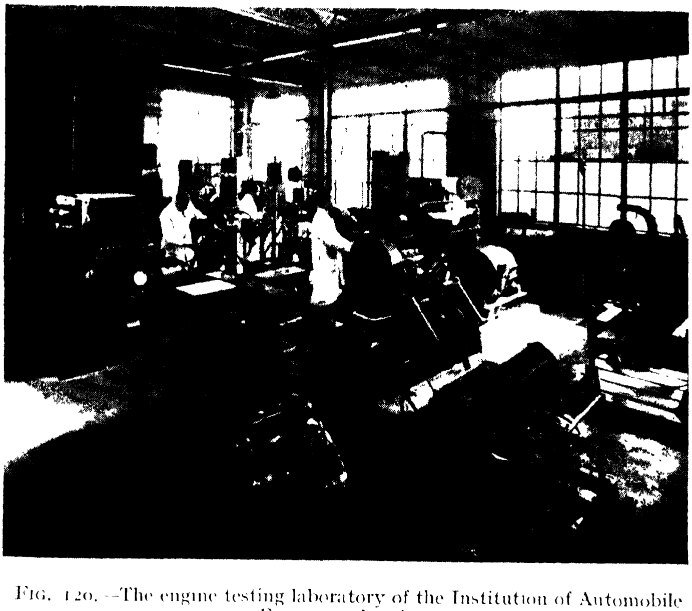


FIG. 120.—The engine testing laboratory of the Institution of Automobile Engineers, London.

[*To face page* 164.

Hydraulic Absorption Dynamometer.—In this type the mechanical energy derived from the engine under test is converted into heat energy, in virtue of the frictional resistance and eddying motion set up in the fluid employed. The general arrangement which finds favour is that of an inner rotating member, or impeller, and an outer fixed or balanced casing. The applied torque of the engine shaft is transferred through the rotor and fluid to the casing, where it is usually measured.

The hydraulic type absorption brake possesses the advantage over the solid friction types described that there is no "snatching," the motion being a relatively smooth one. Further, since the fluid friction is very low at slow speeds and increased rapidly with speed

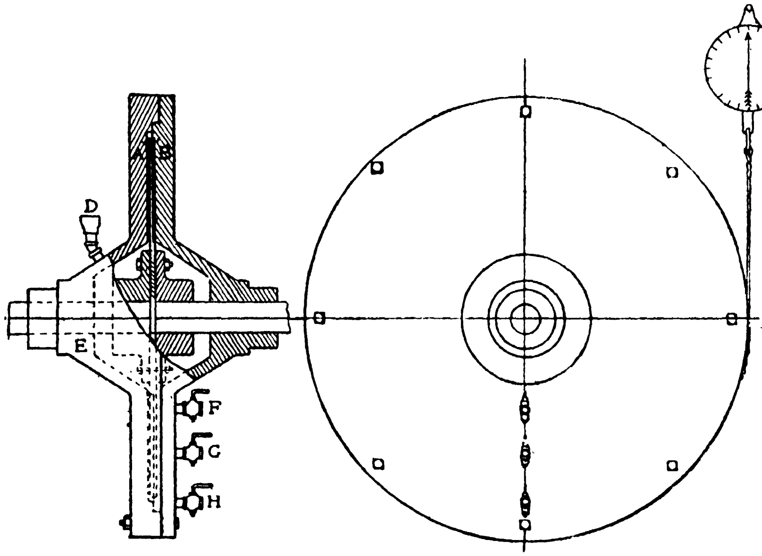


FIG. 121.—Simple form of hydraulic absorption brake.

increment, it is better adapted to petrol engine testing than the solid friction type, in which hand regulation over the speed range is required.

The hydraulic type brake is well damped, and provided the torque-casing friction is not appreciable, and that there is no torque-reaction due to the momentum of the entering and leaving water, its accuracy is sufficiently high for most purposes.

The simple form of water brake shown in Fig. 121, consists of a radial plate having perforations, or projections, this plate rotating within a fairly narrow casing filled with water. The water is circulated through the casing, entering it radially at D, and leaving at one or other of the exit cocks F, G, or H. The rotation of the casing is constrained by means of the spring balance shown on the

right, the scale reading giving the amount of the force required at the radius shown. The product is the torque transmitted, and the horse-power can readily be deduced from the formula given on page 159.

The Brotherhood Brake.—In this type, fluid friction is utilized to absorb the energy. Several perforated annular plates are secured to the casing of the brake, and between each pair a disc rotates with the shaft. The rotating discs run near the fixed plates, but do not touch them. On the introduction of water into the casing frictional resistance is set up between the fluid and the plates. The moment of the frictional resistance between the fluid and the fixed plates is measured by means of a spring balance on the casing torque arm, the casing being free to rotate within limits on trunnions and anti-friction rollers. The water is made to circulate through the casing, and the rate of flow need be sufficient only to keep the temperature down to the desired limits. For example, when absorbing 100 B.H.P., assuming that the temperature of the water supply is 60° F., and that in the casing it is 160° F., about 5 gallons per minute would be sufficient. The water enters and leaves the casing with the minimum of agitation or shock ; for short tests the water in the casing may be used, without continuous flow, there being a constant and definite torque resistance at any given speed, with a definite quantity of water in the casing. It is thus easy to adjust the machine to give a predetermined load curve.

The machine is an accurate one, the only power not recorded being the very small amount of friction between the casing and its supporting roller bearings. The casing rests freely on the rollers, and motion at the supports only occurs when the load changes, and is then extremely small.

The friction of the shaft bearings is included in the readings of the dynamometer, being part of the " absorption " torque measured, so that no error is introduced.

It is claimed that engines under test may be started with the brake set for full load, the torque at the lowest speeds being very small, but rising rapidly as the normal speed is reached.

Multiple Disc Water Brake.—Fig. 122 illustrates a simple form of multiple disc dynamometer which has been employed as an auxiliary brake by the American Bureau of Standards,[1] in connection with the altitude laboratory tests of aircraft engines.

It consists merely of a series of alternatively fixed and rotating perforated steel discs, the intervening spaces being filled with water. There are four water outlets in the casing, each corresponding with a fixed water level. When any one of these outlets is opened and the rate of water flow is adjusted approximately, a constant water

[1] Report No. 44 : " The Altitude Laboratory of the Testing of Aircraft Engines," Amer. Nat. Advis. Committee for Aeronautics.

level is maintained, which is reasonably independent of small variations in the supply pressure. The brake, under these conditions, works very satisfactorily, and is free from the tendency to "drift" towards higher or lower loads with small changes of water pressure.

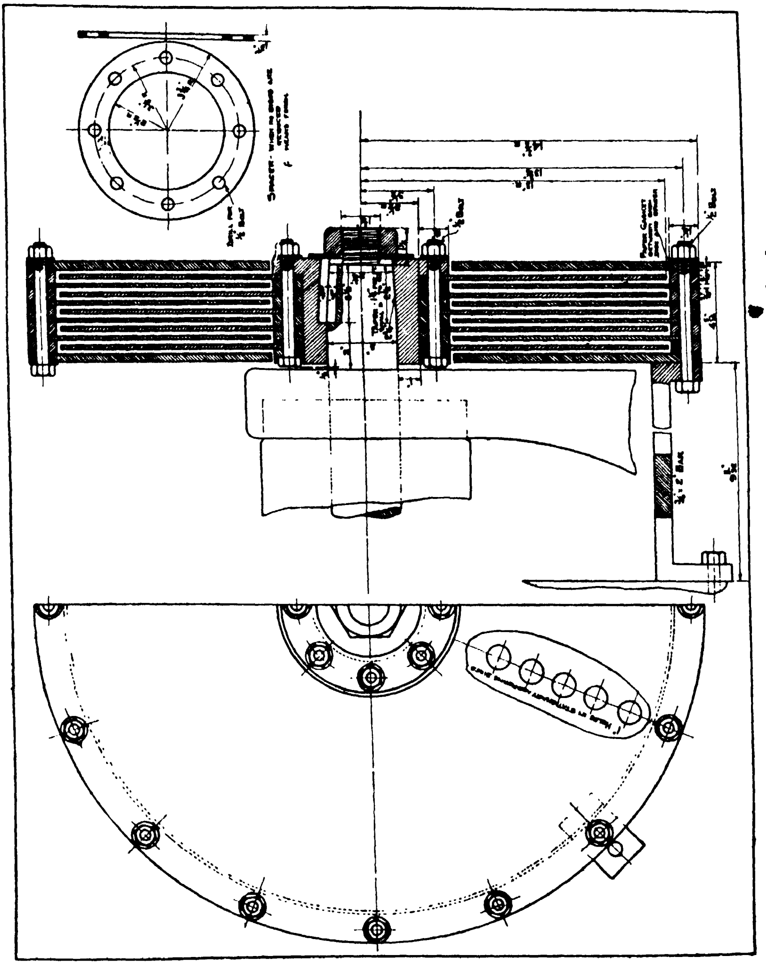


FIG. 122.—The Bureau of Standards water brake.

The water brake illustrated can absorb 400 h.p. at 1800 r.p.m. It was designed as an auxiliary brake to an electric swinging-field one, and its rotor and casing were attached to the corresponding parts of the electric dynamometer.

The Froude Dynamometer.—This now widely-used hydraulic dynamometer was originally devised by William Froude, F.R.S.; [1]

[1] "On a New Dynamometer for Measuring the Power Delivered to the Screws of Large Ships," *Proc. Inst. Mech. Engrs.*, July, 1877.

it has since been developed to a high degree of perfection, and is particularly suited to aircraft and automobile engine testing.

Fig. 123 illustrates the D.P.X. type of dynamometer, in cross-section ; the principle of its action and use can be followed from this diagram.

The main shaft is carried by bearings fixed in the casing (not in external supports). The casing in turn is carried by anti-friction trunnions, so that it is free to swivel about the same axis as the

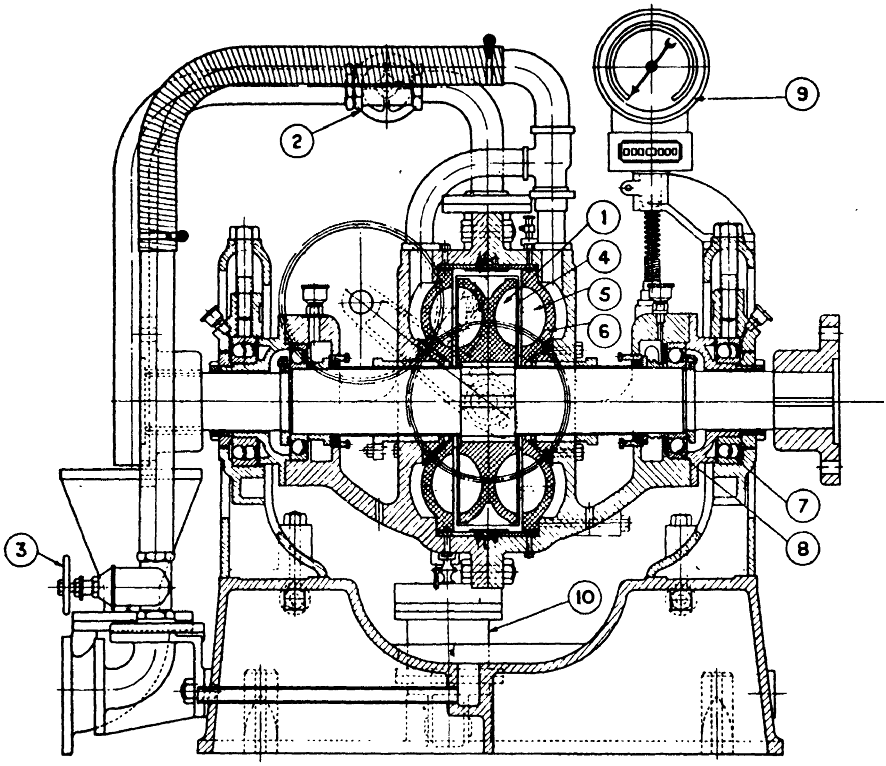


FIG. 123.—Cross-section through casing of Froude dynamometer. (Type D.P.X.)

1. Rotor. 2. Water outlet valve. 3. Water inlet valve. 4. Sluice gates for load control. 5. Water inlet holes in vanes. 6. Casing liners. 7. Casing trunnion bearing. 8. Shaft bearing. 9. Tachometer. 10. Dashpot.

main shaft. When on test the engine is directly coupled to the main shaft, transmitting the power to a rotor revolving inside a casing, through which water is circulated to provide the hydraulic resistance and simultaneously to carry away the heat developed by destruction of power.

In each face of the rotor are formed pockets of semi-elliptical cross-section divided one from another by means of oblique vanes. The internal faces of the casing are also pocketed in the same way.

When in action, the rotor discharges water at high speed from its periphery into the spaces formed in the casing, by which it is then returned at diminished speed into the rotor pockets at a point near the shaft. Thus, the pockets in rotor and casing together form elliptical receptacles round which the water courses at high speed, creating vortices which destroy the power of the engine as quickly as it is developed.

The resistance offered by the water to motion of the rotor reacts upon the casing, which tends to turn on its anti-friction roller supports. This tendency is counteracted by means of a lever arm terminating in a weighing device which measures the torque.

From what has gone before, it will be seen that the forces resisting rotation of the dynamometer shaft may be divided into three main classes as follows :—

1. The hydraulic resistance created by the rotor.
2. The friction of the shaft-bearings, which are usually of the ball-bearing type.
3. The gland friction.

Each one of these forces reacts upon the casing, which, being free to swivel upon anti-friction trunnions, transmits the sum-total of the forces to the weighing apparatus.

The power is calculated from the speed and lever-arm load, from a rational formula of the type considered on page 159, viz. :—

$$\text{B.H.P.} = \frac{\pi \text{DNF}}{33000}$$

$$= \frac{\text{F . N}}{\text{K}}$$

where F = load at end of dynamometer arm and N = r.p.m., K being a constant for the dynamometer.

For example, in the case of a machine of the above type built to British standards, having an arm of 5 feet $3\frac{1}{10}$ inches, $K = 1000$. For a Metric machine having an arm 1432·4 mm. long, $k = 500$. In the ordinary non-reversible type, a small part of the load F is registered by the pointer of the spring balance, the remainder by weights, so that any change in the output of the engine is shown, at once, on the dial. This greatly facilitates tuning up ; slight adjustments to carburettor or ignition, short-circuiting individual plugs, etc., having their effect indicated automatically on the dial of the balance, while the engine is running.

Load Regulation (Sluice Plates).—Referring to Fig. 123, it will be observed that between the rotor and the casing liners are interposed thin metal plates (4) which can be advanced or withdrawn by means of a single hand-wheel. If these plates be moved towards the main shaft they will cut off communication between the rotor

and a number of cups, with the result of diminishing the effective resistance of the dynamometer, and *vice versa*. This means of adjusting the load to suit the capacity of the engine can be utilized while the engine is running, so that in the space of a few minutes a power curve can be obtained over a wide range of speed.

Water Supply.—A supply pipe should be coupled up with the inlet flange, and a return pipe with the outlet flange connection on the dynamometer bedplate. The water should arrive at the dynamometer at a pressure of 15 to 45 lb. square inch, depending upon the maximum speed which will ever be desired. The pipes should be sufficiently large to pass 3 to 4 gallons per B.H.P. per hour when mains water is used, or 6 to 7 when working a continuous

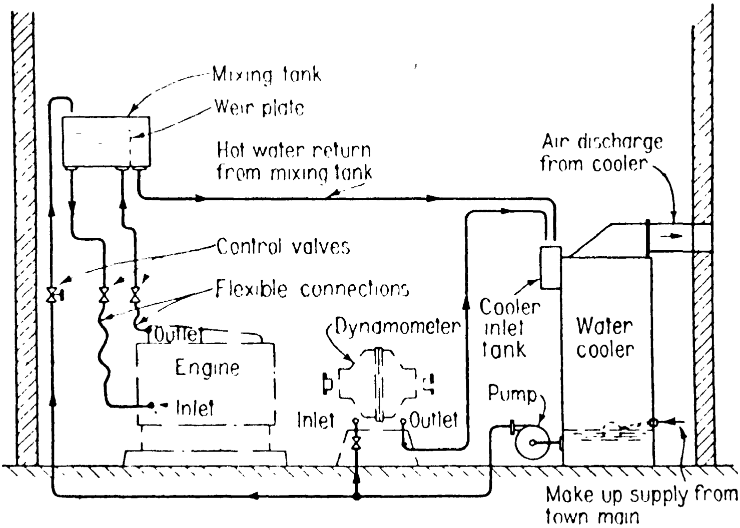


FIG. 124.—Diagrammatic piping layout for test plant with Heenan water cooler.

water-cooling circuit. The water can leave the machine at any temperature up to 140° F.

Violent fluctuations of pressure should be carefully avoided, or guarded against.

A satisfactory self-contained system of water-circulating pump and water-cooler, due to Messrs. Heenan and Froude, is that shown in Fig. 124. This arrangement renders the test house independent of other sources of demand, and saves the cost of a constant running to waste of water from the mains.

Fig. 124 shows a small overhead mixing tank which is frequently used in the engine water-cooling circuit. This mixing tank allows the rate of flow of water through the jackets to be regulated as desired, independently of the rate of cooling water flow through the water

cooler. It is also advantageous when a number of engines is dealt with by one water cooler, as the small quantity of water in the mixing tank enables each engine to be warmed up quickly. The centrifugal pump indicated is chosen so that its head changes only slightly even when its rate of discharge is varied considerably. By selecting a pump having these characteristics the dynamometer or dynamometers can be served with water directly from the pump, the inlet pressure through the machine remaining sufficiently steady for operational purposes even when various test benches are started up or shut down.

Types of Froude Dynamometer.—There are two principal classes of Froude dynamometer, viz. the D.P.X. or D.P.Y. and the S.A. and F.A.

The D.P.X. type is made in small and medium sizes for dealing with high speed engines for motor vehicles.

The D.P.Y. operates on identical principles, except for a slightly different form of weighing gear to cater for higher torques, and is made in larger sizes for dealing with aircraft engines and similar duties.

The torque characteristics of these engines are such that they cannot satisfactorily be determined by certain other forms of dynamometer. Incidentally, these types are applicable to high speed electric motors, steam or gas turbines up to about 4000 h.p.

The S.A. and F.A. types are designed for medium and large sizes of slow-speed Diesel, oil, gas, and steam engines. They are made in various sizes. Two typical ones are the 9000 B.H.P. at 93/150 r.p.m., and the 15,000 B.H.P. at 105/225 r.p.m. dynamometers.

Fig. 127 illustrates a sectional view through an S.A. and F.A. type dynamometer. In this case the main shaft is mounted upon bearings housed in the casing (not in external supports). The casing is supported upon anti-friction trunnions, and is connected up to weighing apparatus, previously described. The power-absorbing pockets and vanes are generally similar in shape to those incorporated in the D.P.X. type of dynamometer, but the vanes in the casing are drilled with air holes in addition to water holes. No sluice plates are interposed between rotor and casing, but the load is regulated by varying the volume of water contained by the power-absorbing pockets. The shape of the pockets is such as to create the formation of eddies in exactly the same manner as in the D.P.X. type, but in the case of S.A. and F.A. type dynamometers, the pockets do not always remain full of water, and the vortices which they contain are free to cavitate under the action of centrifugal force. When working full of water they exert maximum resistance to rotation ; when nearly empty, each pocket contains a vortex consisting merely of a thin film of water coursing around the periphery of the pocket and resistance to rotation is small.

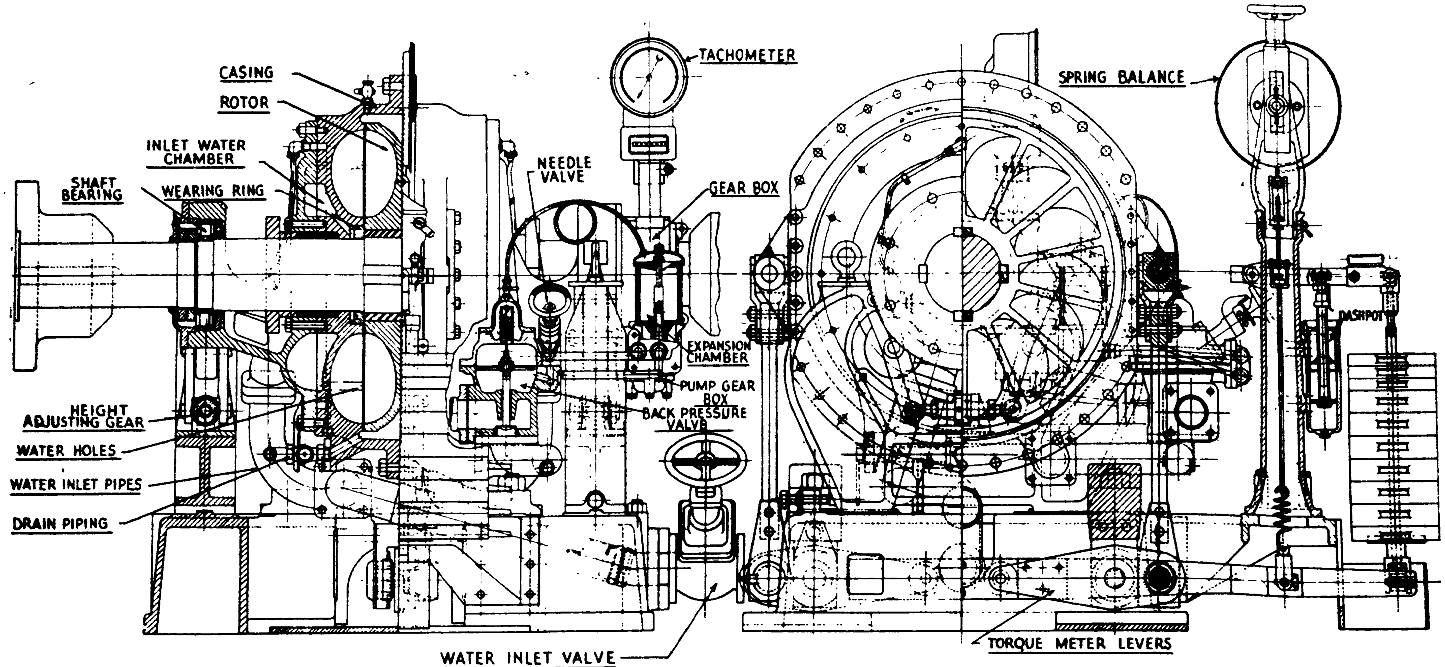


FIG. 127.—The Froude Type S.A. and F.A. dynamometer in side and front elevations.

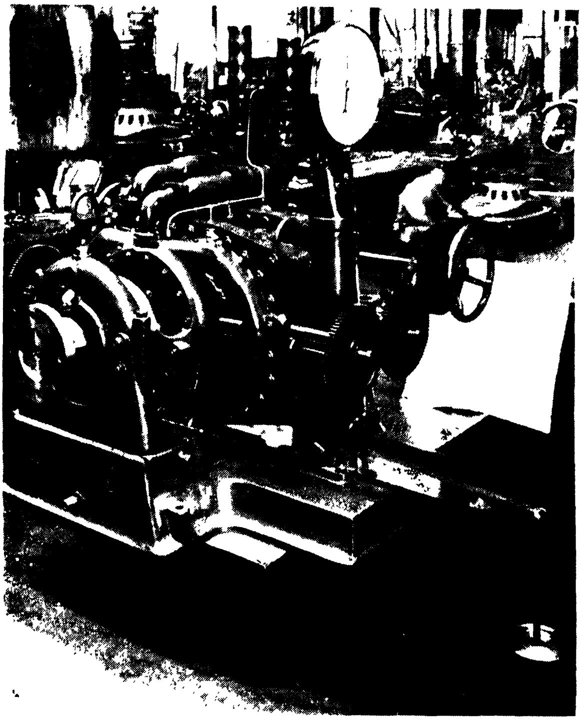


FIG. 125. Froude dynamometer, type D.P.Y. of 3000 h.p. capacity, for aircraft engine testing.

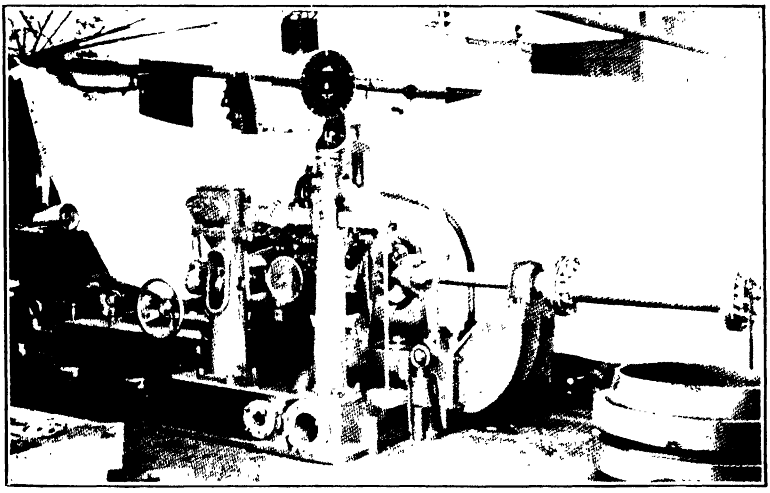


FIG. 126.—Reversible Type Froude dynamometer, 1500 h.p. capacity, with airscrew thrust arrangements, for aircraft engine tests.

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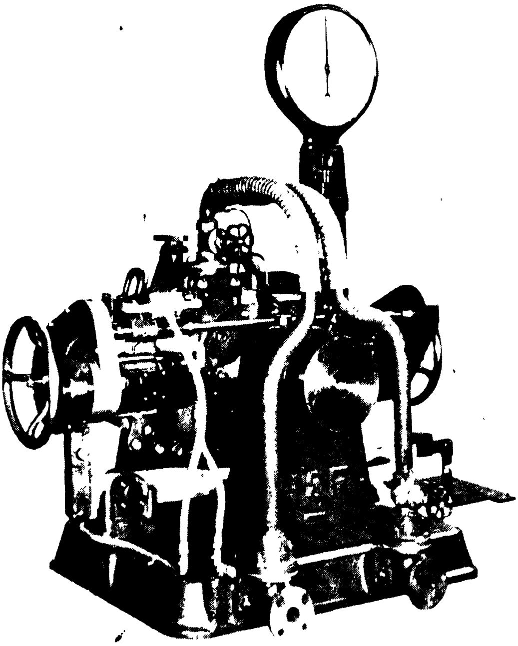


FIG. 128. Froude dynamometer, type D.P.Y.S. 700 h.p., with remote control of load by self-contained electric drive of the sluice gates, push-button operated.

[See page 173.

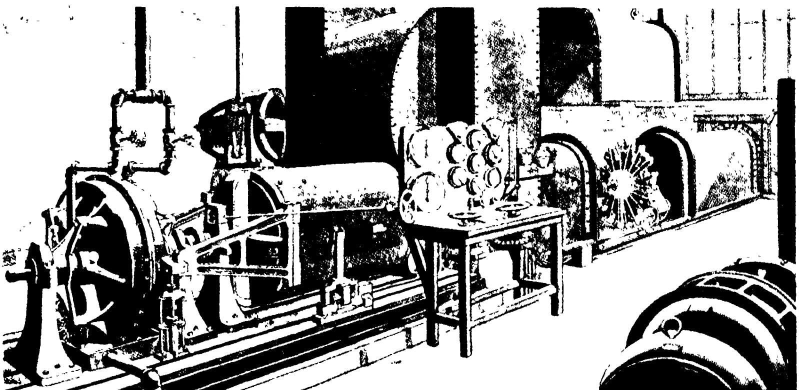


FIG. 130. G.E.C. aircraft engine testing plant, comprising electric and hydraulic dynamometers in tandem.

[To face page 173.

The partial closing of the outlet valve fixed upon the casing increases both the volume of water in the pockets and the load, and vice versa.

Reversible Type.—The Froude dynamometer is also made as a reversible type (Fig. 126), but this is somewhat heavier and more expensive than the other non-reversible standard models.

Running-in Type.—Another type of Froude brake is that designed for imposing a simple load on the engine in order rapidly to bed down the bearings and piston rings.

This type is not fitted with means for regulating or registering the load; but an approximate curve of capacity is given by the makers for each machine, from which the power corresponding with the engine speed is obtainable.

These brakes are noiseless in operation, and are safer than the usual fan type of brake often used for the same purpose.

For general running-in purposes it is usual to choose a brake which will limit the engine speed to about 50 per cent. to 60 per cent. of the normal output of the engine at full throttle.

For example, an engine having a maximum capacity of 75 h.p. at 1800 r.p.m. should generally be run-in at about 1000 r.p.m.; this would be equivalent to 40/45 B.H.P.

Incidentally, the ordinary Froude dynamometer is available in portable form, being fitted with trolley wheels running in a track so that it can be moved from one test-bed to another.

Starting of Engines.—In order to start up engines coupled with water-absorption dynamometers, it is necessary, in the larger sizes, to employ an electric motor driving the dynamometer shaft through a self-disengaging claw coupling, or preferably an automatic free-wheel device.

Remote Control for Hydraulic Dynamometers.—In instances where engines are controlled from sound-proofed chambers the operators are situated some distance away from the dynamometers and remote control of the latter is usually a necessity. In the case of the distantly controlled Froude hydraulic dynamometer the load-control gear is operated by a small electric motor by means of two push-buttons situated in the operator's chamber. Fig. 129 illustrates a typical motorised load-control gear [1] in which the slow-speed shaft of the geared motor is provided with a pinion which engages with the normal gear wheels by which the load-controlling sluice gates of the dynamometer are moved. The motor is of the reversible type, each push-button giving the opposite direction of rotation. The speed of the motor is such as to give a reasonably short time from "full" to "minimum" load positions, whilst permitting small variations of load to be made with ease. A travelling nut operates time-limit switches at each end of the

[1] "The Hydraulic Dynamometer, *Autom. Engr.*, Sept. 1941.

movement in order to prevent over-running. To permit the use of hand-operating gear, if required, the motor pinion can be moved axially to disengage it from the main gears.

G.E.C. Hydraulic Dynamometers.—These dynamometers consist of a specially constructed stator of the torque-reaction pattern, within which one or more rotor discs are arranged. Water entrained within the stator impedes the rotation of the disc or discs and provides a medium for transmitting the torque to the stator. The casing of the latter is provided with a force measuring scale, as in the case of the Froude hydraulic dynamometer, and a similar formula is employed for estimating the horse-power absorbed.

For very high power measurements a combined electric and hydraulic dynamometer unit is provided ; the electric and hydraulic dynamometers are connected in tandem (Fig. 130) on a common base with the rotors and also the stator casings coupled together, respectively.

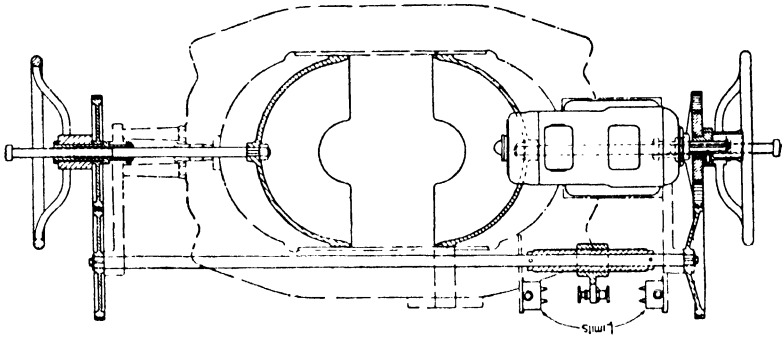


FIG. 129.—Typical arrangement of motorised load-control gear.

Air Absorption Brakes.—This type, which includes also aircraft propellers, depends for its action upon the absorption of power by the churning or air friction of a number of radial blades mounted on a central shaft.

It is now generally conceded that this type of brake is more suited to running-in, and endurance tests, rather than to accurate measurements of power. It has been shown that the horse-power absorbed by a fan brake, consisting of a number of flat plates fixed on radial arms, in the manner indicated in Fig. 131, depends upon the size and shape of the blades, their distance from the axis of rotation, the size and shape of the blade arms, the density of the atmosphere, and the speed or r.p.m. of the shaft to which the blades are attached.

The following formulæ, based upon the results of a large number of tests, are given by J. L. Hodgson :—[1]

[1] " Some Notes on the Fan Dynamometer," J. L. Hodgson, *Proc. Inst. Autom. Engrs.*, March, 1916.

$$\text{H.P.} = \frac{\mu N}{63000}$$

where $\mu = \mu_a + \mu_b$ (the sum of the torques due to the arms and the blades alone, respectively) in in.-lb.

and $\mu_a = 0{\cdot}35 \times 10^{-8} \,.\, \sigma \,.\, N^2 \,.\, r_a \,.\, t \,.\, W,$

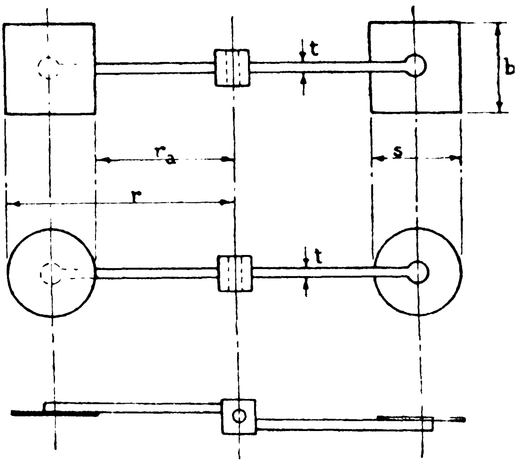
$\mu_b = k \,.\, N^2 \,.\, r^3 \,.\, S^2 \,.\, W.$

The notation employed is shown in Fig. 131, and given below :—

W = Density of the fluid in which the fan rotates in lb. per cubic foot

= 62·33 for water at 50° F.

$= \dfrac{1{\cdot}347 \times \text{barom. press. in inches of mercury}}{(460 + \text{temp. in } ^\circ \text{F.})}$ for air.



Torque (at crankshaft)

FIG. 131.

H.P. = horse-power. N = r.p.m.

S = length of side of square blade, or diameter of circular blade, in inches.

r = external radius of fan, in inches.

r_a = radius of arms in inches, measured to the inside of the blades.

t = thickness of arms in inches.

k = a coefficient depending upon the value of r/s, and the shape of the blade for fans rotating in free space.

σ = a coefficient depending upon the section of the arms.

In the original paper, curves are given with the values of k and σ for various conditions.

The principal deduction drawn from these formulæ is the dependence of the output upon the atmospheric density, which, in turn, depends upon its temperature, pressure, and moisture content.

Further, it has been found that the output is influenced by the presence of air draughts or obstructions, so that for reliable measurements the fan brake should work in an enclosed chamber. The principal disadvantage of the free air type fan brake is its inability to vary the load at a given speed ; in the fixed type this can only be accomplished by stopping it, and moving or changing the blades.

By enclosing the brake in a square or circular casing, however, and providing a variable inlet for the air, or a variable outlet, a wider range of regulation can be obtained.

Fan brakes are generally calibrated electrically, that is to say, with an electric motor of known efficiencies at various speeds, currents, and voltages.

The Walker Fan Brake.—This type, which has been patented by Mr. W. G. Walker, is probably one of the simplest and cheapest forms of dynamometer available, and one which has been very widely used for small electric and petrol motors.

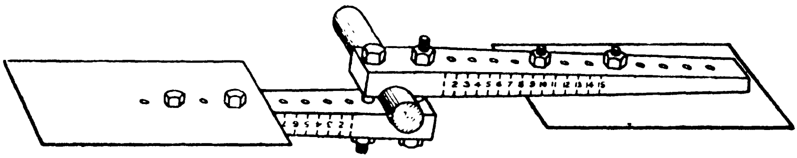


FIG. 132.—The Walker fan brake.

As will be seen from Fig. 132, it consists of a pair of radial arms, which can be bolted to the engine shaft, each carrying a rectangular shape blade, which can be bolted at different radii along the arm.

For each pair of radial arms, and definite size of plate, a calibration curve is given by the manufacturers, from which, and with a knowledge of the engine speed, the horse-power can be computed. A small correction for atmospheric conditions is also employed. Fig. 133 is a reproduction of a typical set of calibration curves for a pair of $8\frac{1}{2} \times 17$-inch plates at different radii, from 1 to 15, along the arms. It will be seen that in this case the horse-power absorbed between 400 and 1300 r.p.m. can, by suitable combinations of radii and speeds, be varied from about 2 up to nearly 60.

The range of sizes in which these fan brakes are provided extends up to about 260 h.p., the smallest giving from 3 to 8 h.p.

The formula employed, when greater accuracy is desired than can be obtained from the calibration curves directly, is

$$\text{H.P.} = k \,.\, N^3$$

where k = a constant depending upon the particular size of brake and the blade position. Tabulated values of this constant can be provided,

and N = r.p.m.

The necessary correction for atmospheric variation is obtained by multiplying the horse-power obtained by the constants, or from the curves, by the following expression :—

$$\frac{B}{t + 460} \times 17{\cdot}4,$$

where B = barometric pressure in inches of mercury,
t = temperature in degrees Fahrenheit.

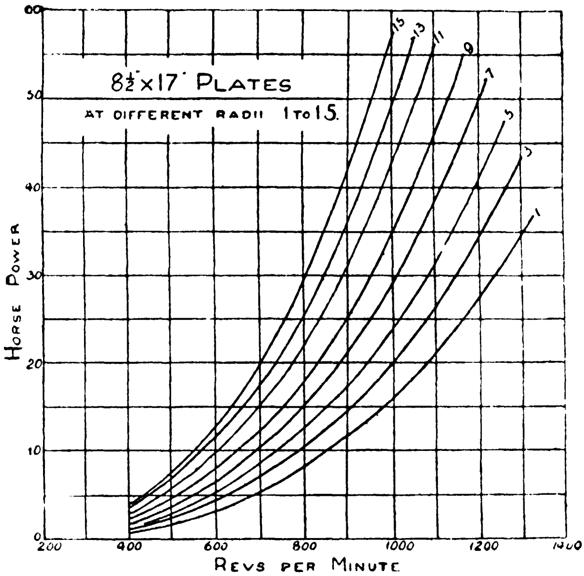


FIG. 133.—Calibration curves for fan brakes.

For the 8½ × 21½ type C fan brake, the following are the values of the constants and speeds :—

TABLE XII

Radius	Maximum Speed, r.p.m.	Constant
1	1400	·000,000,0580
10	—	·000,000,1074
21	1300	·000,000,1965

In the smaller aircraft engine type of fan brake there is a central metal hub which can be keyed to the engine shaft, in a similar manner to the ordinary propeller, with ash arms, and aluminium or steel plates. A pair of engraved plates on the ash arms indicate the radial positions of the blades, which latter are clamped in position by means of two bolts each.

Variable Output Fan Brakes.—One drawback of the fixed form of air brake is that a given fan will only run at one speed when the engine is delivering full power. It is for this reason that much attention has been given to the question of varying the power output for a given speed value.

One method that has successfully been used by French engine manufacturers is to vary the pitch or angle of the fan blades; another method also used was to vary the diameter of the blades. Both methods involve mechanical complications and are not satisfactory for high speed work. The results of some tests made with a variable speed fan dynamometer [1] by the American Bureau of Standards, in which the cylindrical housing of the brake was provided with variable openings, showed that it was possible to obtain a power ration of 5 to 1. In this case the power ratio is defined as the ratio of the power absorbed by the fan at any speed with the outlet open, to that absorbed at the same speed with the outlet closed.

Since the power absorbed varies as the cube of the speed, the speed ratio obtainable in this case was $\sqrt[3]{5}$, or 1·7. Subsequent improvements enabled a speed ratio of 2 (equivalent power ratio of 8) to be obtained.

The enclosed type of fan brake, illustrated in Fig. 134, is also available for engine tests. This design of "*escargot*" or "shell" brake was at one time used largely in this country and in France for aircraft engine tests.

The casing is cylindrical, and the resistance to the revolving blades can be varied during running, by regulating the volume of air passing through. The whole casing is arranged to slide axially, its position being indicated on the top horizontal cross member (Fig. 134). The resistance is a maximum with the outlet and inlet fully open, and a minimum when both are closed.

The fan itself has eight radial blades in this type. In the usual type, with the cylinder at position O completely covering the blades, 5·45 h.p. is absorbed at 1000 r.p.m., and in the position 7 (i.e. an axial movement of 7 inches), the horse-power absorbed is 48·5 at the same speed.

The instrument can be obtained with either 8, 4, or 2 blades, the power output being smaller in the latter case, and under the same conditions.

Another air-dynamometer which employed a centrifugal fan impeller of special design, with provision for varying the amount of power absorbed for utilizing the air blast of this impeller for cooling the engine, was the Heenan-Fell one described in the last edition of this book. It has since been superseded by the Froude

[1] "A Variable Speed Fan Dynamometer," K. D. Wood, Amer. Bureau of Standards, Tech. Note No. 26, Amer. N.A.C.A.

wind-tunnel plant working with a hydraulic dynamometer or a combination electro-hydraulic dynamometer.

Cooling Blast Factor.—In considering the use of this type of dynamometer, it is important to ensure that the cooling blast maintains about the same cylinder temperatures as those experienced when the engine is fitted to an aeroplane and in normal flight. Any over- or under-cooling will evidently give a false impression, both of the power output and of the behaviour of the engine.

The rational method, when using air-blast cooling, therefore, is to actually measure the cylinder temperatures in a few important places and to arrange for these to be the same as those occurring in normal flight conditions.

Electric and Magnetic Absorption Brakes.—In the early days of engine testing, when the available forms of dynamometer were very restricted, it was often convenient to couple the engine under test to an ordinary dynamo, bolted down to the test-bed, and to measure the currents and voltages of the field and armature circuits. Knowing the heating, windage, and other losses of the dynamo at different outputs, its efficiency could be determined, and the actual output of the engine could be computed. Although in the hands of a skilled engineer or physicist, the efficiencies could be determined fairly accurately, there is an element of doubt introduced when this method is more generally used. For this reason it has dropped out of favour and has been superseded by the torque-arm transmission type.

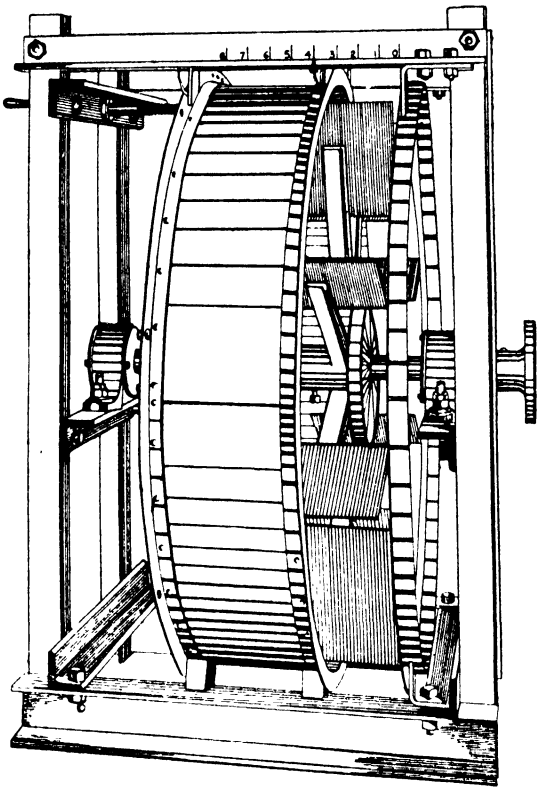


FIG. 134.—The Walker " Escargot," or enclosed, fan brake.

Its one outstanding advantage was that by suitably arranging the electrical connections, the dynamo could be used as a starting motor for the engine, and also for light load motoring tests.

The late Professor Watson employed two similar dynamos, each convertible into a motor on the same test-bed, and was able thereby to apply the Hopkinson test for efficiency.

Eddy Current Dynamometers.—The magnetic type absorption brake has not been used to any extent. Professor Watson made a small eddy-current brake for a 6-h.p. engine, in which a copper disc attached to the crankshaft rotated very close to a series of pole-pieces of electro-magnets, separately excited. Messrs. Morris and Lister have also used the eddy-current testing brake, and a short account of this brake is given in Jervis Smith's book on Dynamometers.[1] In both of the examples mentioned an external source of current was employed; this is a wasteful process, since the engine's output is also wasted. Both types utilized the torque-reaction method of measuring the power absorbed by the brake.

More recently eddy-current dynamometers have been used in the United States for engine tests. Thus the Buick engine test laboratories employ a power absorption equipment having an electric coupling of the "eddy-current" type which permits operating the engine under load at any speed above the synchronous speed of the alternators, i.e. 900 r.p.m. When it is desired to apply loads at speeds below this value the alternators can be stopped and held stationary by means of a brake, and the coupling then used as an "eddy-current" absorption dynamometer.

The Electric Absorption Dynamometer (Swinging-Field Type).—In this now widely used form of dynamometer, the armature is direct coupled, through a flexible coupling, to the engine crankshaft, and the frame carrying the field magnets is mounted either on knife-edges, rollers, or ball-bearings in such a manner that, unless constrained, it can rotate about the armature axis as centre. The torque-reaction is thus measurable by means of a torque-arm with a scale-pan or weights, whilst the current generated in the armature windings can be usefully employed for other purposes. In most of the existing types, the field coils are separately excited, the dynamos being of the shunt-wound direct-current type. Load and speed are regulated by providing a variable resistance in the field circuit.

This type of dynamometer, and of which the G.E.C., English Electric, Heenan, Vickers and the Sprague types are examples, not only provides a convenient, accurate, and steady brake, but also enables the dynamo to be converted readily into a motor for starting purposes, and for engine loss motoring tests. It has a wide speed range, and gives a heavy braking torque at low speeds. If the electric power given out is not required for useful purposes, it can

[1] Constable & Co., London. See also "Mechanical Testing," vol. ii, Batson and Hyde (Chapman & Hall, Ltd.).

conveniently be absorbed by means of resistance coils placed in the open, or by a water resistance.

Utilizing Brake Power Output.—Although it is not always practicable to use the electrical output of absorption brakes for works purposes, owing to the fluctuating nature of the tests, yet for endurance and also preliminary test runs this power can be usefully employed.

Recently the Ford aircraft engine works has evolved a method of utilizing the power output of engines which are given a preliminary run of several hours each. Previously this power was wasted owing to the engines having been fitted with power-absorbing airscrews; moreover, the noise was objectionable.

To save this power, sixteen Westinghouse 1250-K.V.A., 720 r.p.m., synchronous generators were installed. Each generator is connected to an engine through a hydraulic slip coupling, so arranged that when the engine speed exceeds 720 r.p.m., the generator runs at a constant speed and delivers power to the plant alternating-current bus. Sufficient power is recovered in this manner to supply a large percentage of that needed for the manufacture of the engines.

Typical Examples.—An early example of this "swinging-field" type of dynamometer was that used for the official tests, at the National Physical Laboratory, in 1911, of aircraft engines in connection with the Patrick Alexander Competition.[1] This apparatus had a swinging frame carrying the field magnets, which rocked upon knife-edges, about the armature axis. The torque-arm was provided with an autographic device for obtaining the power curve.

Professor Watson[2] was finally led to adopt the "swinging-field" method in preference to the direct-coupled fixed generator one. The arrangement employed is shown in Fig. 135.

The engine was connected to the armature shaft through a flexible coupling of the Zodel-Voith type. The current generated was absorbed by a series of air-cooled resistance coils; the field coils were excited from a separate source, namely, the power mains.

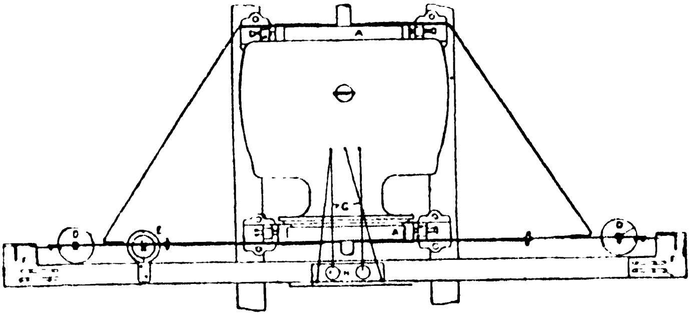
The armature circuit resistance could be varied by means of a rheostat, so that the load and speed were capable of regulation. The apparatus could also be used for motoring the engine. Referring to Fig. 135, the motor casing, carrying the field coils, was provided with two cast-iron rings A, which rested upon pairs of rollers B, made of hard steel. The cast-iron was not sufficiently hard enough, and so strips of spring steel were used for the roller contact surfaces. A transverse beam with scale-pans D at each end was used to measure

[1] *Vide* Reports on the Tests of Petrol Motors in the Alexander Motor Prize Competition, 1911, H.M. Stationery Office.
[2] *Proc. Inst. Autom. Engrs.*, November, 1912.

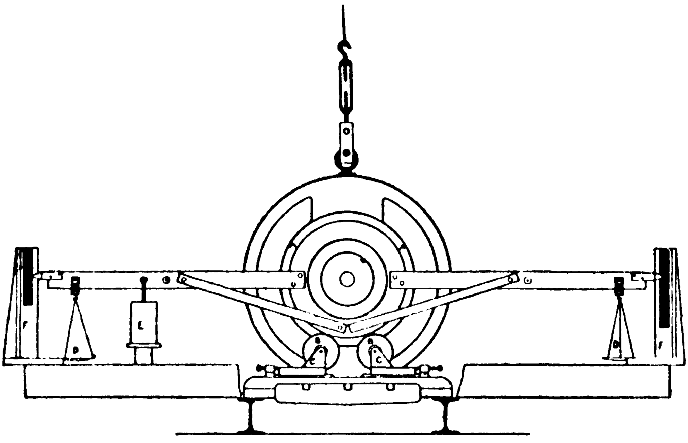
the torque in either direction, whilst an oil dash-pot E was provided in order to damp out vibrations.

Most of the weight of the motor casing was taken by means of a knife-edge in the eyebolt at the top.

This knife-edge was carried by a wire attached to one end of a pivoted beam, the other end of which carried a counter-weight, so that the motor casing was held in a state of stable equilibrium.



PLAN.



FRONT ELEVATION.

FIG. 135.—The Watson swinging-field torque-arm electric dynamometer.

This method enabled multiples of the pound on the scale-pans to be used, while fractions could be read off by means of pointers moving over the scales F. The dynamo was of about 9 kilowatt capacity, and the arrangement shown enabled a torque of one pound-foot to correspond to a deflection at F of 3·6 mm. It was possible to read the scale to within one or two-tenths of an inch accuracy.

The swinging-field or cradle type of dynamometer was also used

by Professor S. P. Thompson in 1884, and by Dr. Drysdale in 1905. Ricardo has employed the same form of dynamometer for most of his research and test work. These were manufactured by Messrs. Vickers, Ltd., to the design and specification of the former. Figs. 19, 28, and 29 illustrate one of these swinging-field type dynamometers installed in the test laboratory.

The dynamometer consists of an ordinary shunt-wound 220 volt direct-current inter-pole generator, and at both ends the armature shaft is extended to take flexible couplings to enable two different engines to use the same dynamometer in turn. The complete machine is carried on ball-bearing trunnions in cast-iron standards, which are securely bolted down to the test-bed. To each side of the field casing is attached a light torque-arm, which is carefully balanced. Each torque-arm terminates in a bracket carrying a pointer, which registers with a scale carried on a bracket mounted on a steel standard which is fixed solidly to the test-bed or floor. At the end of each torque-arm are carried respectively the weight-pan or spring balance, and a small oil dash-pot, the connections to which are of the Skefko ball-bearing type. The dash-pot is about an inch larger, in internal diameter, than the damping-plate on the torque-arm connection, so that the metal parts do not touch.

Readings can be taken at each end of the torque-arms, as these are identical. Thus, power readings can be taken from one arm, and friction losses when motoring, at the other, the direction of rotation being reversible by reversing the shunt field.

Each dynamometer set is provided with a switch panel (as shown in Fig. 19) carrying (*a*) a double-pole main switch; (*b*) a change-over switch in the armature circuit, to enable the armature to be connected up either to the power circuit of the test-house for the motoring tests, or to a series of resistance coils, when generating; (*c*) a starter; (*d*) a two-pole field switch; and (*e*) two field rheostats, one having about 25 stops, and giving a rough adjustment of the field strength, and the other having about 100 steps or graduations which are arranged in series between each step of the coarse rheostat. A very fine adjustment is thus possible over a wide range.

The shunt fields are separately excited from the main power circuit, the torque and speed being adjusted by the two rheostats previously mentioned.

The average load on the torque-arm, in the case of a small car engine, is about 30 to 40 lb., and this device is sufficiently sensitive to detect a change of load of less than one-half an ounce, thus giving a working accuracy of about 1 in 1000. This degree of accuracy is particularly advantageous, not so much in power tests as in the friction loss tests, where the engine components are successively dismantled, and the motoring losses measured in turn.

The dynamometer described is not only sensitive, but is dead-

beat, showing no tendency to oscillate under load. Suitable resistances for such dynamometers can be made from coils of galvanized iron wire suspended vertically on wooden frames placed in the open air. Frames measuring about three feet by two feet, carrying about 12 lb. of this wire wound into spirals, will accommodate about 10 h.p. absorption each. This type of resistance very quickly attains its working temperature, and its resistance does not vary afterwards.

In connection with the field coil excitation, it is fairly easy to obtain a range of torque or speed variation of 2 : 1 without changing the armature current value.

It is preferable to employ a battery circuit for the field supply, in order to maintain the field current uniform.

Notes on Electric Dynamometers.—The type of dynamometer selected depends upon the engine test routine. In some instances the engines are run in under their own power ; in others the electric dynamometer is used as a motor to drive the engines. Further, it is convenient to be able to convert the dynamometer to a motor for engine starting purposes.

The torque-reaction or swinging frame dynamometers are preferable to the fixed frame pattern which require the use of ammeters, voltmeters, or kW/H.P. meters for power measurements and an accurate knowledge of the efficiencies of the machine under different load and speed conditions. When the torque-reaction type is used it is an advantage to be able *to lock the stator* or swinging portion to the frame when no horse-power readings are being taken in order to reduce the wear on the weighing arm mechanism ; this is done in the case of the English Electric Company's electric dynamometers used for internal combustion engine tests. For routine engine tests a brass dial spring balance gives sufficiently accurate readings, if calibrated in pounds, but as its spring tends to oscillate an absolutely steady reading is not always possible. For precision tests the steelyard with sliding weights should be employed in preference to the spring balance.

In some instances the spring balance is used in addition to the steelyard to give rapid readings and also to act as a damper to take up sudden shocks. It is, however, an advantage to fit an oil dashpot type of damper with the moving baffle member attached to the torque arm or stator casing and the fixed oil container to the frame. The dynamometer should be connected by some form of flexible coupling to allow for slight misalignment with the engine.

The design of the armature of the dynamometer requires special consideration. Thus, the English Electric Company has found that in order to keep the diameter of the armature as small as possible, it has been found advantageous to employ higher temperature rises than those common in the industrial field ; for instance,

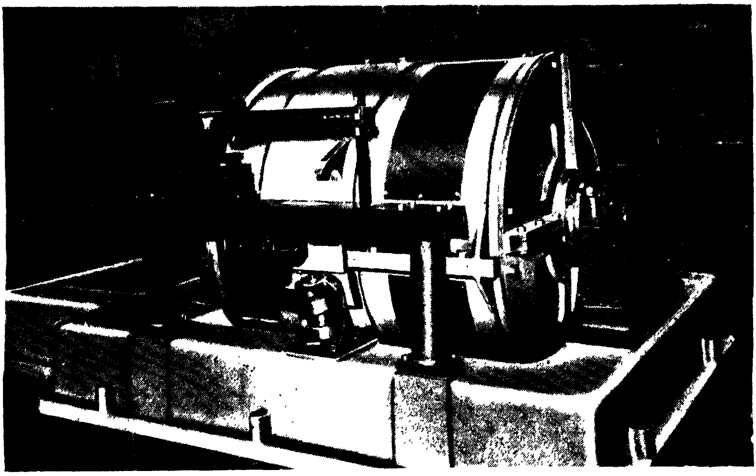


FIG. 136. English Electric Company swinging frame type dynamometer with weighbridge load device

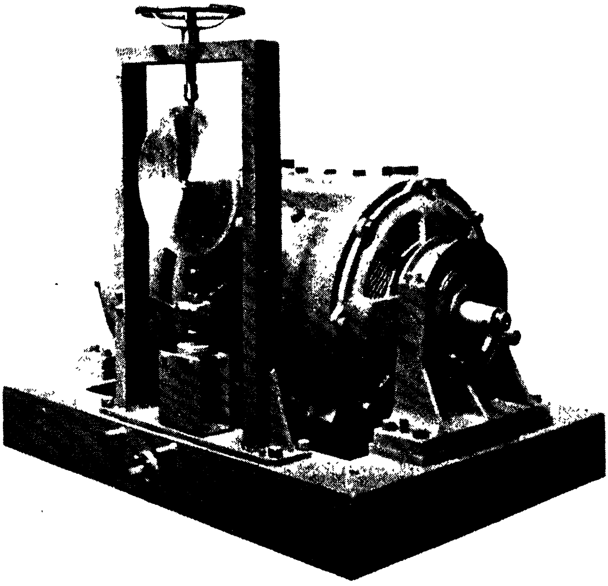


FIG. 137. English Electric Company swinging frame dynamometer with spring balance load device

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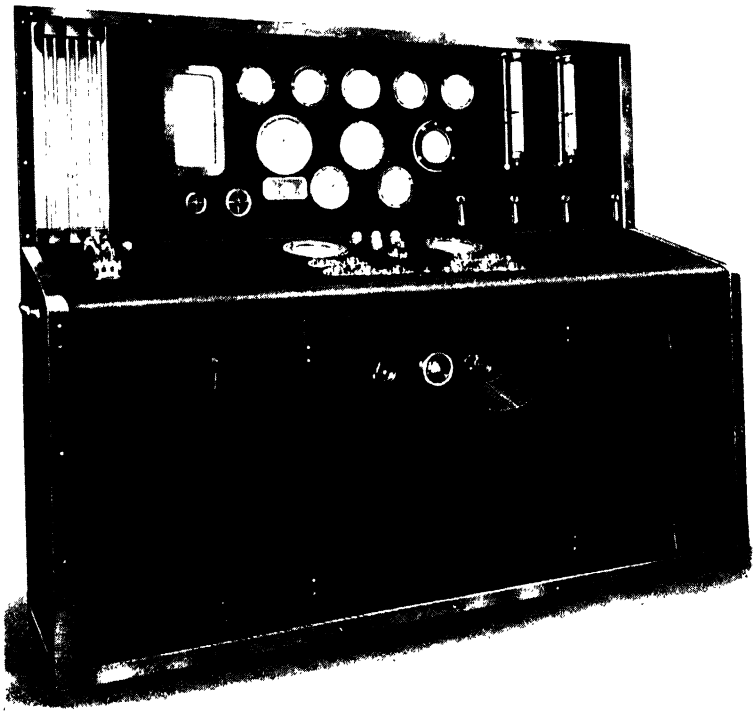

FIG. 138. The English Electric Company combined engine test and dynamometer control desk for supercharging test measurements

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it has been found possible to design a machine with an output of 300 h.p. at 3600 r.p.m., or 150 h.p. at 6000 r.p.m., in a single frame with a temperature rise not exceeding 105° C. by resistance. This temperature rise is nothing unusual, however, as it has been standard for traction motors for a considerable number of years.

The field system of the dynamometer should be accurately balanced on its own trunnions, and the balance should be such that if the frame is moved slightly from its normal position, it will remain in the new position.

It is important that the ventilating air in the dynamometer is suitably controlled by baffles, so that there is no disturbing torque reaction on the frame from this source.

For swinging-frame dynamometers it is obviously necessary to supply flexible connections for the main and field leads and, in the case of the main connections, these can be conveniently made by means of a number of very thin copper strips or of copper braid.

In the case of aeroplane engines, *the supercharger is usually tested separately* from the engine and, since the test routine consists of running the supercharger over a wide range of speed, a variable-speed D.C. motor forms a convenient means of doing this.

It is frequently required, however, to take accurate torque readings for this test, and therefore the swinging-frame dynamometer is again used. The tendency in aeroplane engine construction is towards testing as many individual parts as possible, all of which necessitate a variable-speed drive, and electrical machinery provides a convenient means of doing this.

Generally speaking, the D.C. type of dynamometer provides a more flexible speed and load control than the A.C. one and it has been more widely employed; the increase in use of variable speed A.C. motors of various types may, however, result in increasing use being made of the A.C. pattern dynamometer.

The Heenan Electric Dynamometer.—One form of this testing equipment has been designed, primarily, for the running-in and power testing of production engines in the works; it has a series of automatic signals and controls. The method adopted is first to motor the engine around so as to bed down the working surfaces until the friction is reduced to a given value. The running-in is begun first at a low speed and with a relatively heavy torque. As the "stiffness" of the engine diminishes the speed is increased and the torque reduced.

The engine is then allowed to run light under its own power to enable the pistons and rings to run-in under heat conditions. The throttle is then opened and endurance tests or power curves are taken.

The testing plant utilizes the well-known electric motor-dynamo combination, in which the frame of the electrical machine is free to

rotate about the armature shaft axis, but is constrained by a spring-balance arrangement ; the latter is used to measure the force at the end of the frame torque-arm. Fig. 139 is a sectional view of the Heenan electric dynamometer, the various parts being indicated by the lettering on the illustration. There are two sets of ball and roller bearings, one for the armature shaft in the casing and the other for the casing in the fixed pedestal bearings.

When used for running-in purposes the electrical machine is arranged to work as a motor. When the engine is actually working under its own power the motor becomes a dynamo and delivers

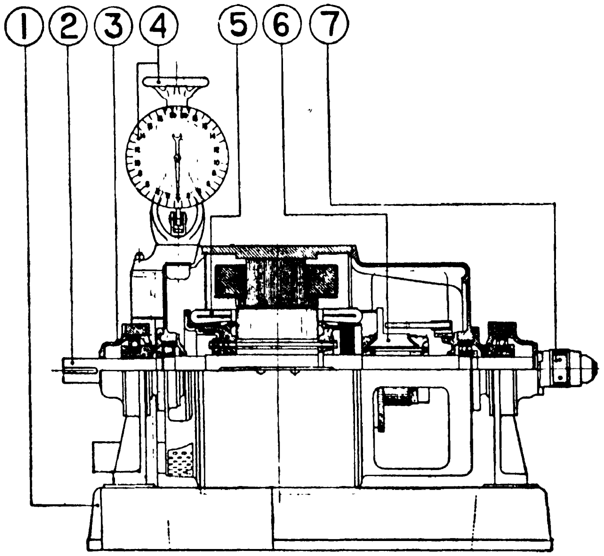

FIG. 139.—Sectional view of Heenan electric dynamometer.

1. Fixed motor base-plate.
2. Armature shaft.
3. Motor casing bearings.
4. Spring balance and control wheel.
5. Armature windings.
6. Commutator.
7. Driving end of shaft.

electrical energy. If the engine speed is high enough this energy can be returned to the main electric power supply of the works, or laboratory, so that the greater part of the power given out by the engine is recuperated.

When the engine is running at low speeds under its own power, as when making power tests at low speeds, the dynamo will not generate line voltage ; it is then changed over to resistance loading.

During both the running-in and light load running periods, the friction or power can be read off the dynamometer torque-arm dial.

Any tendency to overloading the engine is met automatically by the engine cutting its ignition off and coming to rest.

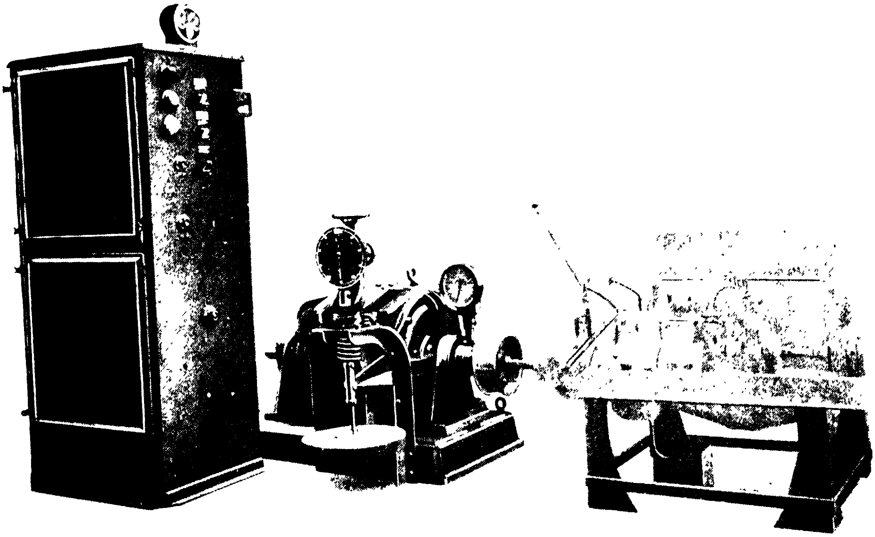

FIG. 140.—A general view of the Heenan electric dynamometer testing plant.

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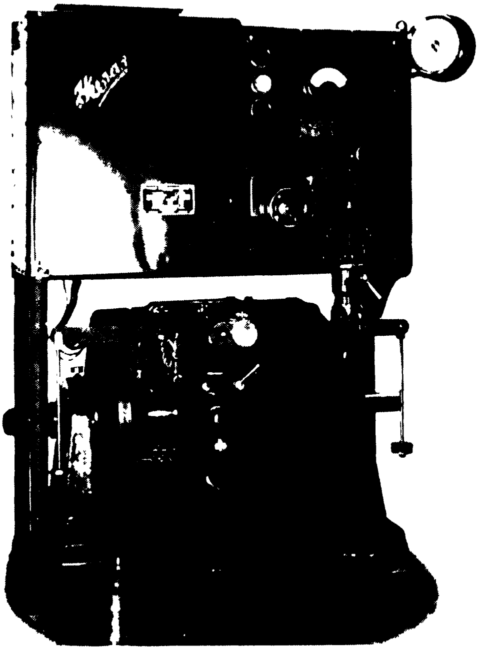

FIG. 141. Typical single frame torque-reaction Heenan electric dynamometer

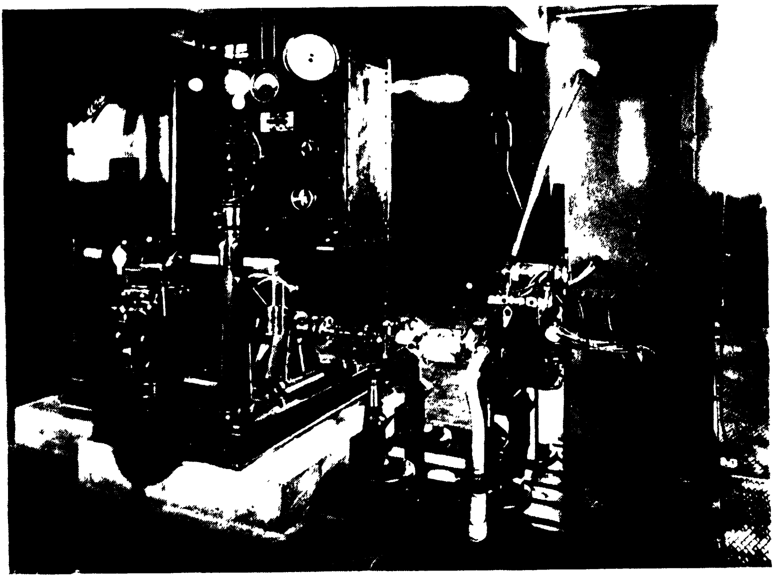

FIG. 142. Heenan electric dynamometer for production testing. (Messrs. S. I. M.C. A., Paris.)

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Various types of engine can be tested with the same plant. The running-in torque which is suitable for a large engine is normally too heavy for a smaller one. Adjustment is, however, provided by which the torque may be modified by turning a switch and effecting a rapid internal adjustment in the control panel.

The testing panel and measuring instruments are shown in Fig. 140. The test procedure adopted for mass-production engines is as follows :—

The engine having been mounted in place on the test-bed, the successive operations and indications are :—

(*a*) Set all controls to zero and close isolator. Complete plant is at " safety " and will not rotate until—

(*b*) Start button is pressed. Engine is driven at slow speed and red lamp lights. Red lamp remains lit until internal engine friction has fallen to its predetermined value. When red light disappears—

(*c*) Rotate master control handle to positions 2, 3, or 4 in slow succession. When indicator 4 appears, blue lamp lights, indicating that engine is ready to run under its own fuel. At this stage running-in is complete.

(*d*) Admit fuel and open throttle. Engine pumps power back to line. It should be run first on light load and later at gradually increasing speed and load by adjusting throttle and regulators 6 and 7.

(*e*) Testing for endurance and power curves can be carried out by adjusting throttle and regulators to give desired speeds and loads.

(*f*) To test at low speed, operate a control to reduce speed. Close throttle. The dynamometer controls are then set to load the dynamometer against the resistances.

(*g*) Adjust speed precisely by regulator. Speed can be reduced in this way down to a low value at full throttle.

From the preceding remarks it will be evident that this testing plant can be operated by semi-skilled assistants requiring little knowledge of the theoretical side. It is thus possible to test engines rapidly in the works without the usual laboratory equipment and skilled operators.

The calculation of the horse-power from the torque-arm balance reading W and the tachometer speed N r.p.m. utilizes the well-known relation

$$\text{B.H.P.} = \frac{W \times 2\pi RN}{33{,}000}$$

where R = radius of torque-arm in feet.

In the Heenan electric dynamometer the torque-arm R is often made 5·251 feet in length so that the formula for calculating the horse-power becomes

$$\text{B.H.P.} = \frac{W \cdot N}{1000}.$$

In dynamometers built for metric measurements a similar plan is followed. Thus, in a metric machine having an arm 1432·4 mm. long the formula becomes

$$\text{B.H.P.} = \frac{W \cdot N}{500}.$$

In certain cases the principle of torque-reaction measurement of power is replaced by the use of an electric meter, calibrated direct in B.H.P., to indicate the power developed or absorbed. This method, while not so inherently accurate, is sometimes of a sufficiently close approximation, and is frequently used when quantities of the same size of engine have to be tested; but the makers invariably employ for accurate work torque-reaction weighing gear constructed on the same principles as those of the Froude hydraulic dynamometer.

The same firm also manufactures electric dynamometers for *high-speed and high-power tests*. In this case there are two electrical machines mechanically coupled. Under certain conditions these can be coupled electrically in series; in other conditions, in parallel.

This form of dynamometer is in wide employment in laboratories and research departments, for the testing and development of new engine types. It is of high accuracy, one part in 1000 being quite normal, and similar torque measurement gear is used in single-frame high-power Heenan dynamometers as employed in tests on large petrol and Diesel engines, which do not run at extreme speeds, in the laboratory and the production test-house.

It is an interesting feature of all Heenan torque-reaction dynamometers that they indicate direct the actual nett B.H.P. entering or leaving their shafts. All internal losses, including the B.H.P. absorbed by internal fans or other air-cooling devices, are automatically added to or deducted from the readings (as the case may be) according to whether the machine is absorbing or developing power. This is inherent in the design of the dynamometer, and involves no special mechanisms and no particular action by the operator; it obviates the necessity of using correction curves.

In cases in which the powers to be absorbed are outside the capacity of electrical machines, the Heenan electric dynamometer is frequently coupled in tandem with a specially arranged Froude hydraulic dynamometer, which absorbs the excess. In this case also, losses such as those in the Froude dynamometer do not

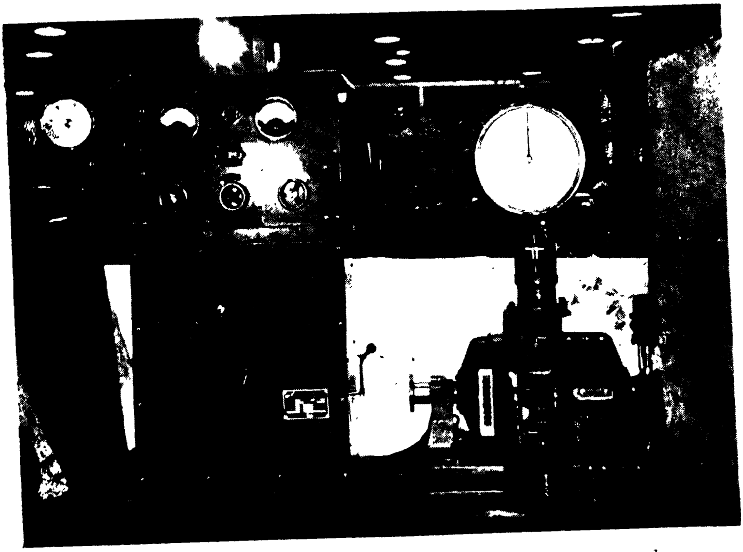

FIG. 143.—Small Heenan electric dynamometer with wide speed range, operated under Ward-Leonard control.

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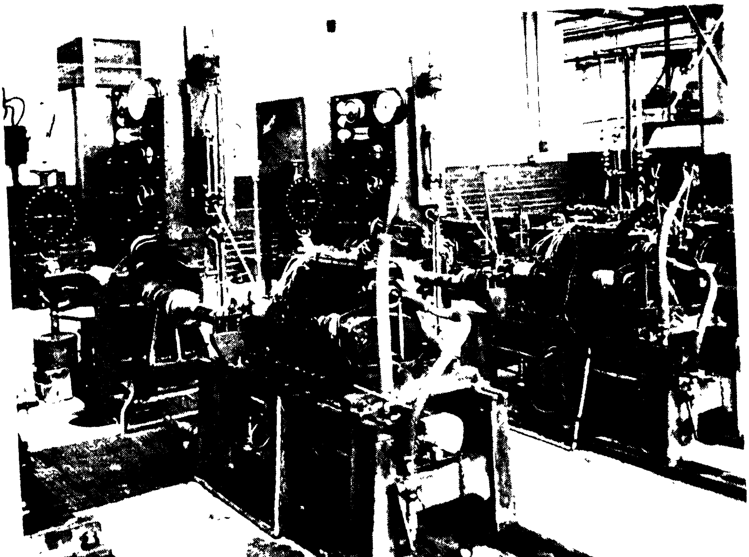

FIG. 144. Battery of Heenan electric dynamometers for production testing. (Morris Commercial Cars Ltd.)

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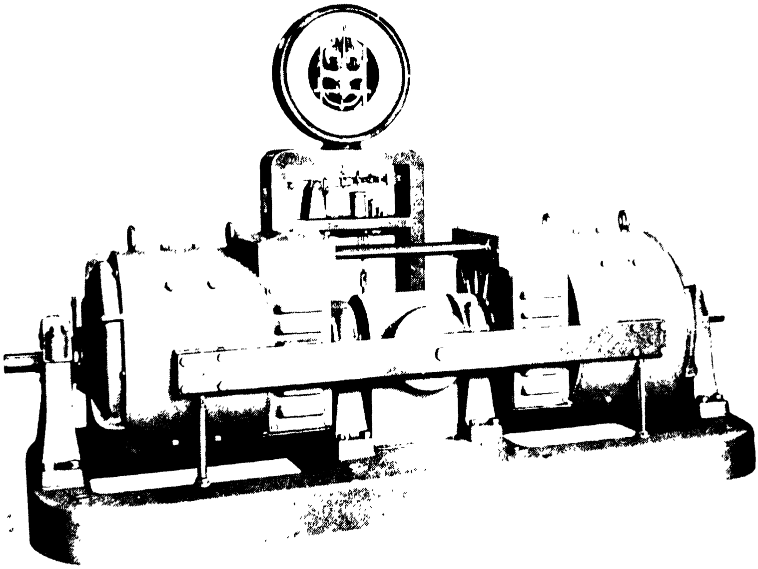

Fig. 145.—G.E.C. engine testing plant, comprising two D.C. generators coupled together and an Avery balance for torque measurements.

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appear in the torque readings, when the electric portion is developing power, but are included when the assembly is absorbing the engine output; this is achieved simply, by correct fundamental design of the torque reaction weighing gear.

The G.E.C. Electric Dynamometers.—These machines operate on the torque reaction method and are designed to operate either as motors or absorption generators in either direction of rotation. The engine to be tested is coupled to the armature shaft through a flexible coupling. Both the armature and stator are mounted on low friction bearings, the stator or casing being free to revolve except as restrained by the measuring scales.

The horse-power transmitted to the armature of the machine when used as a generator is calculated from the usual torque-r.p.m. formula, namely :—

$$\text{B.H.P.} = \frac{\text{r.p.m.} \times \text{lb. pull on scale}}{K}$$

where K is a constant which is usually arranged to be a whole number for each size of dynamometer, e.g. 2000, 3000 or 4000.

The G.E.C. dynamometers [1] vary in size from 4 to 1000 H.P. with maximum speeds up to 5000 r.p.m., but wide ranges of speed above the rated values can be obtained by the use of suitable control equipment. The latter is designed to permit adjustment of torque and speed by the independent control of armature and field current.

There are two different types of control panel available, namely, the standard and fully-automatic ones.

The standard control panel is arranged for operation by one man and furnishes protection for the operator to the engine under test and the dynamometer. The panel contains a voltmeter and ammeter; circular handwheels in the centre for control of the stator (or field); large dial switch for controlling the rotor current, connected to an external rheostat in which the electrical energy developed by the dynamometer is absorbed; circuit-breaker to prevent overload; a field reversing switch for reversing the direction of rotation of the commutator and two contactors (one for line and the other for load resistance). The fully-automatic control panels include all of the features just mentioned and are particularly suitable where space is limited or remote operation is required.

Apart from internal combustion engine tests, these dynamometers have been used for testing transmissions, brakes, tyres, belts, fans, and blowers, centrifugal pumps, waterwheels, motor chassis and steam turbines.

Production Car Engine Tests.—The G.E.C. dynamometers have been employed for production tests of car engines before leaving

[1] General Electric Co. Ltd., Kingsway, London, W.C. 2.

the factory, special block-testing dynamometers being used for this purpose. A typical application is that used for Austin engines of the 10 h.p. and 12 h.p. class (Fig. 146). Twenty-six G.E.C. block-test sets are used in connection with these engines, the pre-war production rate of which was 2000 per week. The testing sets are grouped in two lines of 12 while the remaining two are situated separately and are designed for the special purpose of testing sports engines with a maximum speed of 4500 r.p.m.

Each set comprises a D.C. machine (which can be used either as a motor or a generator), a direct coupled tachometer, and a control pillar housing the necessary control equipment.

The engines, as soon as they are assembled, are brought to the testing department on an overhead conveyor track. On arrival the engine is lowered on to one of the test-beds, according to the size of the engine. Water, oil and petrol supplies are then connected up and the engine itself coupled to the testing machine. These operations are completed in a few minutes by means of carefully designed fittings. Petrol and oil are fed to the engines from a central supply, the consumption of petrol averaging 3000 gallons per week. Five special generators are installed to provide ignition current, while cooling water, maintained at normal working temperature, is circulated by pump from reservoirs to all test-beds.

Running-In Tests.—Before the actual tests are made, the testing machine is made to drive the engine at a low speed in order to run it in. After a suitable time the ignition of the engine is switched on and the engine starts to drive the machine, which now acts as a D.C. generator, feeding current back into the supply mains. The engine is then run up to higher speeds to obtain normal working temperatures, the generator automatically absorbing the engine horse-power which is indicated on a horse-power meter. Up to this point the engine has been running for two hours.

The speed is next reduced to a low value in order to make adjustments. The engine is then tested over the whole speed range at full throttle, and the electric power output is again returned to the mains. During this test the output and speed of the engine are respectively and accurately shown on an ammeter (calibrated in horse-power) and an electrical tachometer.

The maximum output figure must reach a given minimum value before the engine is passed.

For the fine speed adjustment necessary, the testing machines had to be of the D.C. type, but as the power supply at the works is A.C., a 500 h.p. 500 r.p.m. synchronous motor-generator set is installed, converting the A.C. into D.C. at 220 volts. The D.C. testing machines thus take their supply from, or feed back into, the D.C. side of the motor-generator.

Each of the 24 D.C. testing machines is designed for a speed

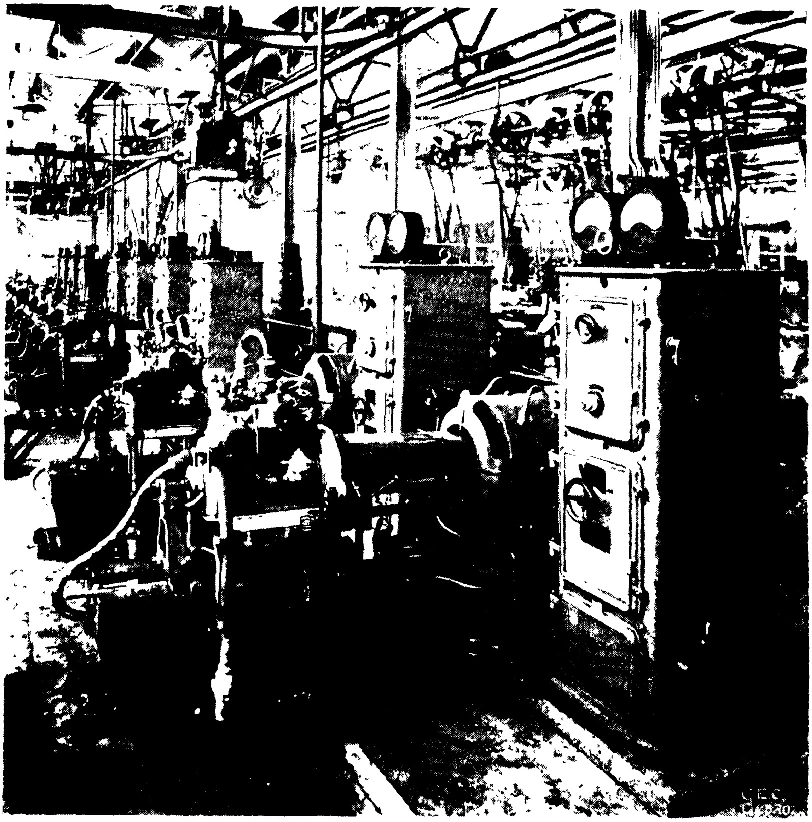

FIG. 140.—Part of the Austin Motor Company's testing department for automobile engines.

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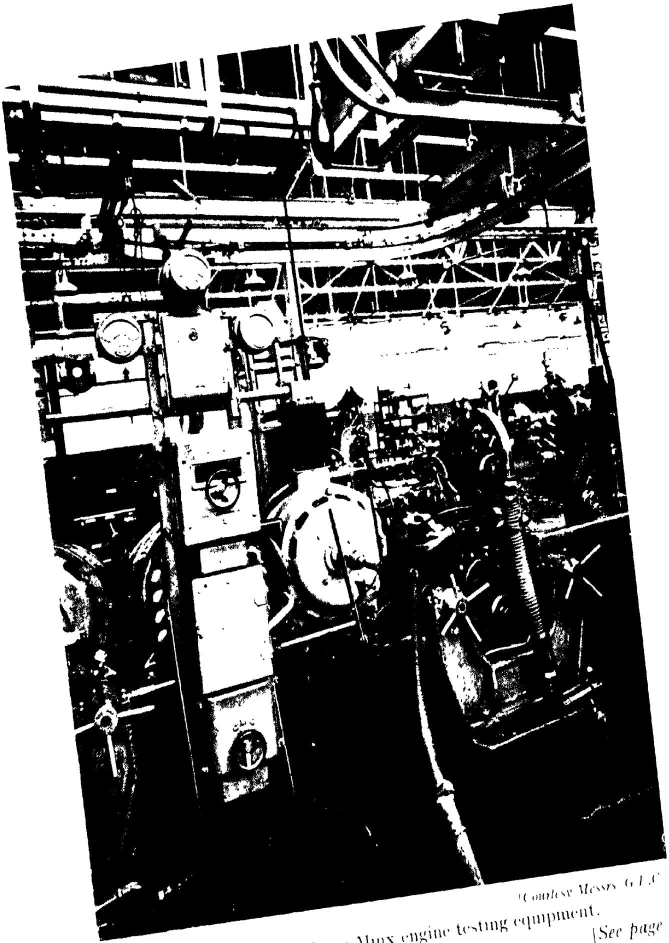

[Courtesy Messrs. G.E.C.

FIG. 147 Hillman Minx engine testing equipment.

[See page

range on load of 1000/3200 r.p.m., and is controlled from an adjacent control pillar. This contains a double pole isolating switch in the base, a double pole ironclad line contactor, a push-button operated face-plate starter, a main shunt wheel regulator, together with a vernier shunt field regulator for giving fine speed regulation. Overload and overspeed relays are also provided, while emergency stop push-buttons are installed at convenient positions, which operate a dynamic braking contactor. A tumbler switch for controlling the engine ignition circuit, an ammeter calibrated in horse-power, and a speed indicator complete the equipment embodied in the control pillar.

Interlocks are provided to render the sets safe for use by unskilled operators.

The machine is started up as a motor by means of a hand-operated starter. It will then drive the engine for running-in purposes at the desired speed. By switching on the ignition of the engine and opening the throttle the machine automatically acts as a generator. Speed variation and load are controlled by the main and vernier shunt regulators and are shown on the speed indicator and H.P. meter.

To stop the set the operator returns the starter to the "stop" position, thereby tripping not only the main contactor, but also the engine ignition circuit. This precaution is necessary for the reason that were the main contactor to be tripped alone, all load would suddenly be taken off the engine which would race away. In cases of emergency the operator can push a button mounted close to the engine and the machine is brought to rest very quickly by means of dynamic braking.

The tripping of either the overload or overspeed relays also brings the machine to rest and an indicator operates to show which relay has functioned. The relays can be quickly reset by the operator.

The two high speed testing sets function in a similar manner to those described. The maximum speed, of course, is higher and has entailed certain modifications to the design of the machines. These are fitted with particularly heavy shafts in order to raise the critical speed well above the maximum running speed, while the commutators are reinforced with steel rings shrunk into position over micanite insulation. This prevents movement of the copper bars and ensures proper current collection under all conditions. A resistance is incorporated in the control system so that the machines can be run at low speeds on load and then changed over to the line.

A further difference is that instead of an ammeter calibrated in horse-power as used on the other sets, a patented compensated wattmeter is used so that the horse-power of any engine within the range tested can be indicated accurately irrespective of the varying speed losses in the testing machine.

Sprague Electric Dynamometer.—The Sprague electric dynamometer [1] is very widely employed for petrol engine and other test purposes in America. It originated in 1906, since when several hundreds of these dynamometers have come into use in laboratories, works, and test-houses.

The dynamometer is the same in general principle as those previously described, the torque reaction being measured on the oscillating field magnet casing, which rocks in ball-bearing pedestals.

The field current is supplied from a separate source, at either 115 or 230 volts; it is intended for direct-current supply. In special cases the dynamometer may be made self-exciting, but its speed range under these conditions is limited, and, except for routine or production tests, separate excitation is considered essential. The power required for excitation of the field is fairly small, but if the dynamometer is required to act as a motor, a large source of supply is essential. Sprague dynamometers vary in size from a nominal 1 h.p. at 1000 r.p.m. up to a nominal 1000 h.p. at 1000 r.p.m. with a series of intermediate sizes. These dynamometers are also available for speeds of 1500 to 4000 r.p.m. The 50 h.p. model is suitable for engines having a maximum torque of 50 to 200 ft.-lb., whilst the 400 h.p. machine is applicable to engines with a torque capacity of 800 to 1600 lb.-ft.

For very large engines, two or more dynamometers can be coupled in tandem, or, as we have already seen, an additional water brake can be used as a supplementary source.

Referring to Fig. 148, which illustrates a typical Sprague-type machine, the torque reaction of the field casing causes a pull in the right-hand link C, which, in turn, causes the link D to act on the spring balance, and to register the amount of the pull. The cross-over beam B is provided with knife-edges at the centre and extremities; the centre pivot is adjustable vertically. A weighing beam above the spring balance enables the larger weights to be measured accurately, whilst the balance gives an indication of the weight, or pull in the link C.

The field casing can be locked for using the dynamometer as a rigid motor, by means of the hand-wheel shown on the left.

For measuring the motoring torque in such cases as those of engine losses, there is a second inverted knife-edge in the middle of the link D, which transmits the torque effort to the balance.

The dynamometer is arranged to take direct current for motoring purposes, as when starting engines, at 115 or 230 volts. For friction horse-power tests, the supply of direct current available should be at least 25 per cent. of the maximum rated capacity.

The control of the load and speed extends over a wide range, a

[1] Manufactured by the General Electric Co. of America.

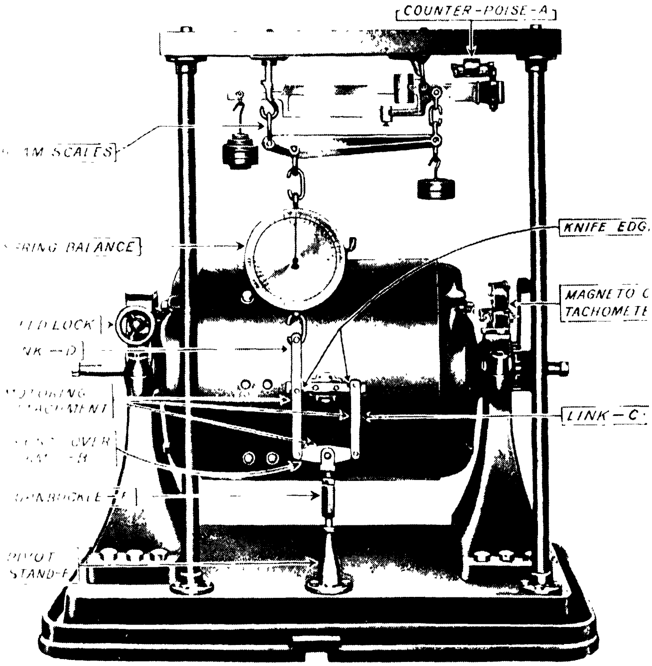

FIG. 148.—The Sprague electric dynamometer (swinging field type).

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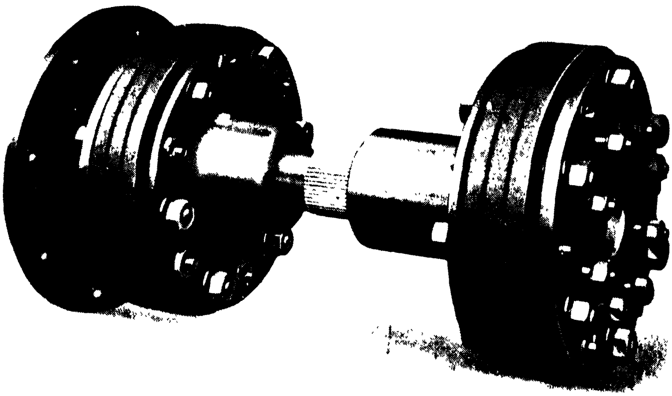

Fig. 149. A convenient form of flexible coupling used on Froude dynamometers
See page 193.

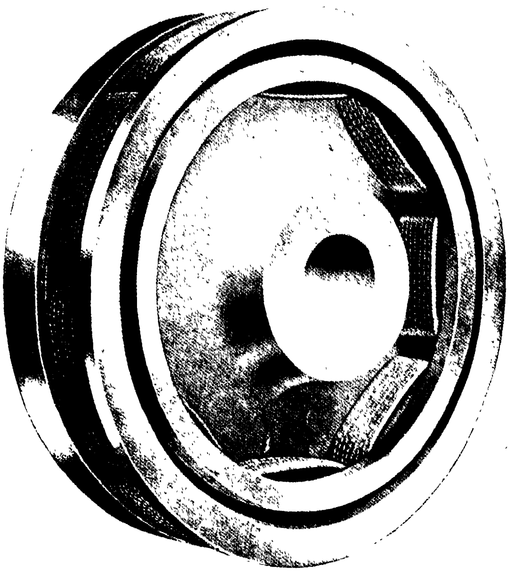

Fig. 150. The Zodel-Voith coupling.
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dual type rheostat being provided in the field circuit for the finer adjustments, and a single dial segment type of rheostat in the armature circuit for the larger load variations. With these two controls a very wide range of speeds and loads can be obtained.

Large Output Electric Dynamometers.—The electric absorption dynamometer can be made in very high output sizes for testing the largest aircraft engines of the present day and also projected future types.

What is claimed to be the biggest electric dynamometers are two 4000 H.P. combination cradle-units made by the American Westinghouse Electric Company for testing aircraft engines. Each consists of one 800 H.P. direct current unit operating at 700 to 2000 r.p.m. and one 3200 H.P. inductor-type unit operating over a similar speed range.

General Considerations.—In connection with the " swinging-field " type dynamometer, it is essential that the electrical leads to the machine shall be sufficiently flexible as not to cause any appreciable resistance, otherwise the measured torque will be less than the true value. For this reason the cables are usually wound helically just above the machine. Professor Watson, in order to minimize this source of error, employed mercury cups on the stator casing, the electric cable ends dippin into these for contact purposes.

Some authorities consider the ball-bearing support for the stator inferior to the knife-edge method ; thus Dr. Drysdale found that there was a possible error of 5 per cent. with ball-bearings, the sensitivity being also reduced, in comparison with knife-edges ; the latter, which are universal upon testing machines, will carry a load of 5 tons per linear inch satisfactorily.

To reduce the frictional torque due to the stator weight still further, the stator is sometimes partly suspended, in the manner described on page 182.

Concerning flexible couplings, a number is now available, of which the Hardy flexible fabric disc and the Zodel-Voith types are commonly employed.

Fig. 149 illustrates the former type, whilst Fig. 150 shows an example of the latter type. Both are free from any metallic connections, or wearing surfaces, so that they are silent, flexible, and of long life.

The Z.V. type shown consists of four parts, an inner and an outer disc, with suitable bosses, a rim bolted to the outer disc, and an interlaced belt of leather or fabric connecting the outer and inner rims. The outer disc can be removed without unlacing the belt, so that either shaft can be removed with a minimum of trouble.

Fig. 151 shows a type of cardan shaft, with flexible couplings at its ends, used by Heenan & Froude, Ltd., and suitable for

transmitting the power of a large aircraft engine. The joint consists of a pair of flanges each having a ring of overhung bolts engaging a semi-flexible disc by which the forces due to torque are transmitted from one ring of bolts to the other. If stress permits, the flexible material may consist of laminated fabric, otherwise good results have been obtained from the use of thin steel plate, heat treated. In some cases the surface of the plate is pressed into a wave forma-

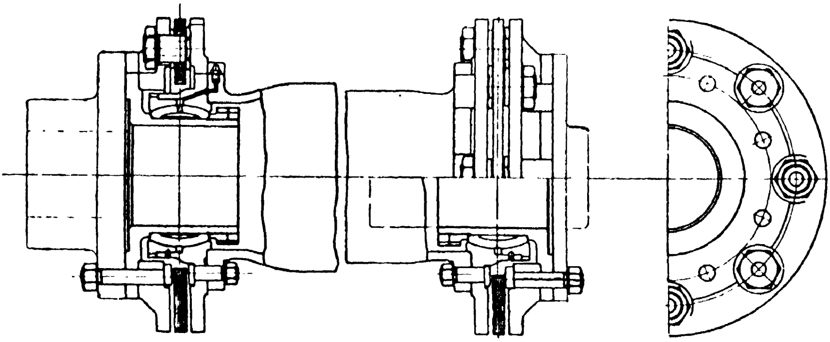


FIG. 151.—Cardan shaft with flexible couplings.

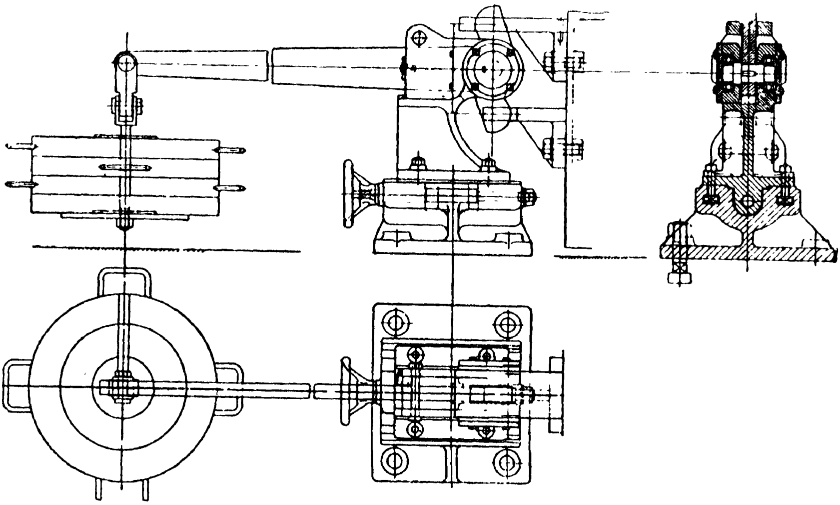


FIG. 152.—Dynamometer for combined power and thrust measurements.

tion in order to give increased flexibility against misalignment. The spherical self-aligning seatings for the flexible joints are capable of transmitting a heavy thrust so that the engine can be subjected to thrust tests while driving the dynamometer.

Combined Power and Thrust Measurements.—A method of combining power measurement and thrust measurement suitable for aircraft engines, employed by Heenan & Froude, Ltd., is

shown in Fig. 152. The dynamometer is carried by a cradle mounted upon a roller-bearing trolley which is connected to a bell-crank lever. Dead-weights applied to the long arm of the lever generate a thrust similar to that which would be developed by a propeller during flight. The dynamometer shaft and pedestals, and the engine cradle, are arranged to take the reaction from this thrust.

Supercharger Tests.—The testing of aircraft superchargers is an important branch of research and routine work requiring the use of a special type of dynamometer capable of giving accurate readings of the power required to drive the supercharger over a wide speed range.

Whilst the single speed gear-driven type supercharger has been widely used, it has now been replaced largely by the two-speed pattern. The impellers of these superchargers are driven at high speeds, namely, from 15,000 to 25,000 r.p.m., and the impeller-drive units must be capable of withstanding high acceleration stresses such as those that occur when the throttle is suddenly opened or closed. It is not possible, owing to space considerations, to describe the types of superchargers in use or performance and test data; for a full account of these subjects the reader is referred to the footnote reference on page 18. The standard Air Ministry tests for superchargers include an acceleration test in the course of which the acceleration time is measured and a given time interval is allowed for the supercharger to reach its maximum speed from rest.

A modern aircraft engine supercharger may absorb from 6 to 8 per cent. of the power developed by the engine itself, at ground level, in the case of engines having rated altitudes up to 15,000 feet. In the case of a 1500 h.p. engine the geared supercharger will require from 90 to 120 h.p. to drive it at its normal operational speed, whilst for altitudes of 20,000 to 30,000 feet the power required has been shown to be from 12 to 20 per cent., i.e. 180 to 300 h.p.

Fig. 153 shows a Heenan electric dynamometer installation for testing high speed superchargers of modern aircraft engines. It is of the double-ended type combined with two gear-boxes giving different speed ratios. Only one gear-box is in use at a time; each gear-box steps up the speed of the motor, while the dynamometer is electrically connected to a Ward Leonard type of motor generator; this arrangement gives a wide and accurate control of speed.

At the high speed end of each gear-box a steel flywheel, enclosed in a sheet-steel casing to avoid inaccuracy due to windage, transmits the power of the motor to a multi-plate friction clutch by means of which the output shaft of the dynamometer is driven. The supercharger is connected to the output shaft by a light and accurately balanced flexible coupling. Normally, the friction clutch remains engaged throughout the tests, except for acceleration tests when

it is first disengaged and the flywheel is brought up to a predetermined overspeed. The supercharger being ready to start, the clutch is engaged suddenly, accelerating the supercharger up to top speed

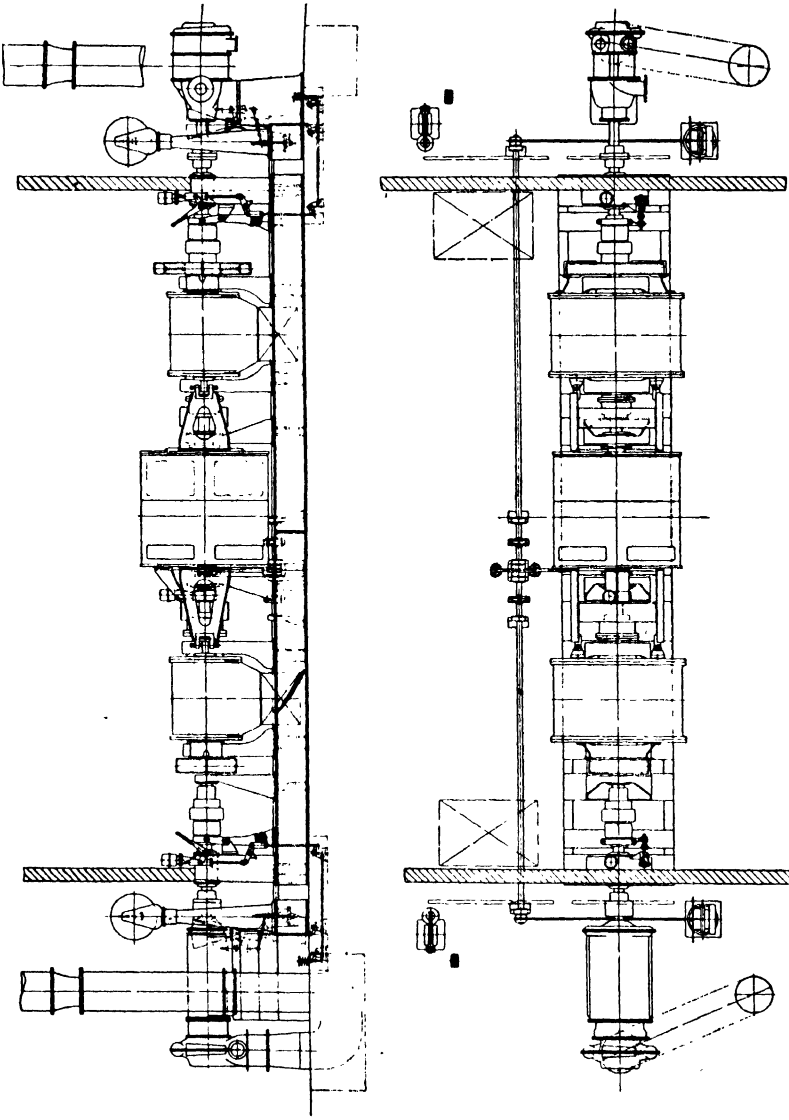


FIG. 153.—Typical layout of Heenan electric dynamometer for testing superchargers.

in a period of time which depends upon the slipping torque of the clutch; provision is made in the designs to adjust this torque, the value of which is measured on the weighing apparatus of the dynamometer. To facilitate control of accelerating time, a variable

dashpot is connected to the throw-over lever of the clutch. The control stations containing all the instruments relating to the dynamometer and supercharger performance are partitioned by a vapour-proof wall from the dynamometer house.

The English Electric Company's supercharger testing plant is referred to on page 185.

Bristol Supercharger Tests.—Before a "Bristol" supercharger is assembled into an engine, it is thoroughly tested as a unit and its performance determined. The "blower test rig," to give the test plant its factory name, consists of a bed-plate on which are mounted an electric motor and a dummy engine crankcase to which the supercharger unit can be attached. The 100 h.p. motor is of the variable speed type and is mounted in trunnion bearings so that the field system may turn about the shaft axis. Thus the torque exerted at any moment may be measured by means of a spring balance attached to the end of a lever fixed to the casing of the motor; the product of torque and speed, divided by a constant, gives the horse-power being put into the drive of the supercharger.

On the air intake there is a calibrated nozzle, across which a mercury U-tube manometer is connected, the drop in pressure across the nozzle, with due corrections for temperature, being a measure of the quantity of air passing. Between the nozzle and the carburettor intake there is a regulating valve, and there is another valve on the final discharge pipe from the supercharger to the outside atmosphere. U-tubes are provided also at the carburettor intake and on the discharge side of the blower, while thermometers show the temperature of the air before and after it has passed the blower.

After running light for a short time in order to see that the blower is up to the required standard of silence and so forth, the speed is increased to correspond to the normal rated speed of the engine, and a two hours' endurance run is started. During this run observations of pressure and temperature are taken at frequent intervals, the inlet and discharge control valves being adjusted as may be necessary to maintain the required conditions of load.

Finally, a calibration curve is taken at uniform speed by varying the quantity of air passed, and from this the all-round performance of the supercharger can be compared with the standard to which it must conform before being accepted. From the observed horse-power exerted by the driving motor and the weight and pressure rise of the air dealt with, the mechanical efficiency of the unit may readily be calculated, and this again must conform to a rigorously applied standard.

Supercharger Calculations.—In connection with the performances of air compressors at various heights as calculated from tests under ground level conditions of pressure and temperature

it has been shown that the performances may be represented as a function of the two variables

$$\frac{W}{\sqrt{P_0 \cdot \rho_0}} \text{ and } \frac{n}{\sqrt{T_0}}$$

where W = weight of air flowing per unit time. P_0 = intake pressure (abs.). ρ_0 = intake density. n = speed of rotor. T_0 = intake temperature (abs.).

Thus, supercharger pressure ratios and efficiencies may conveniently be plotted as ordinates using values of $\frac{W}{\sqrt{P_0 \rho_0}}$ as abscissæ, for constant values of $\frac{n}{\sqrt{T_0}}$.

Many of the results given in the Aeronautical Research Committee's Reports are expressed in this manner.

The calculations employed in connection with official tests on superchargers are given in Appendix No. IV.

Transmission Dynamometers.—The electric swinging-field type of dynamometer might be termed a transmission one, if the engine were designed to drive permanently an electric generator or dynamo the electric output of which could be utilized for some specific purpose. Dynamometers of the transmission type are not used to any appreciable extent in high speed internal combustion engine work of the type coming within the present scope, so that the reference to this form need be a brief one only. Any dynamometer which enables the power output to be read off whilst the engine is performing its allotted duty, and which does not absorb power, comes into this category.

If the engine be mounted on ball, or roller bearings, in such a way that it could, unless constrained, rotate about its crankshaft axis, the torque reaction could be measured by the torque-arm method previously described, but in this case the arm would be attached to the crankcase or cylinder block. The engine torque at any given speed could thus be measured without interfering with its proper working. This constitutes one type of transmission dynamometer. It is not now used to any extent, if at all, owing to the heavy bearing pressures on the torque reaction bearings, and to the difficulty of arranging the exhaust and carburation connections.

The method was used, however, in connection with the German aeroplane engine trials of 1912.[1]

There is another class of transmission dynamometer which depends upon the interposition of a flexible torsion member between the engine and the power absorption unit, such that the torsional de-

[1] "Dynamometers," F. J. Jervis Smith (Constable & Co.).

flections of this member can be measured in a convenient and simple manner whilst the plant is at work.

The Ayrton and Perry,[1] Denny,[2] Hopkinson-Thring,[3] Föttinger,[4] Amsler,[5] and Geiger [6] transmission dynamometers all belong to this class.

The principles underlying the construction of this class of instrument depend upon the torsional properties of a length of shafting, or of a resilient member.

If a shaft of length l inches and diameter d inches be fixed at one end, and if a torque of magnitude T lb.-in. be applied at the other end, the following relation holds :—

$$T = \left(\frac{\pi . C . d^4}{32 . l}\right)\theta$$

where θ = angle of twist and C = modulus of rigidity for the material of the shaft in lbs. per square inch.

For a given length of shaft, it will be seen that, within the elastic limit of the material the angle of torsion θ varies as the applied torque T. Hence, by measuring θ it is possible to deduce the torque T.

Applied to a rotating shaft the same relation holds, for the effect of the transmitted torque will be to cause a slight twisting of the shaft. If this angular deflection is measured, the torque can be obtained.

It has been shown that if T_m is the mean torque transmitted

then $$\text{B.H.P.} = \frac{2\pi . N . T}{33000}$$

so that the horse-power being transmitted at any moment can therefore be computed, or by a suitable choice of scales and read off direct at any speed.

Hopkinson-Thring Torsionmeter.—It is also possible to obtain optical or pencil records of torsional angle variations in certain types of torsionmeters. In the Hopkinson-Thring torsionmeter, which has been fairly widely used for determining the horse-power transmitted by turbine and other shafting, the apparatus, which includes a pair of concentric sleeves, is fixed to a short length of the shaft under test, and the torsional angle read off a scale by means of an optical system.

[1] " Applied Mechanics," by John Perry (Cassell & Co., London).
[2] *Proc. Inst. Naval Architects*, March, 1907.
[3] " Mechanical Testing," vol. ii, Batson and Hyde (Chapman & Hall, Ltd.).
[4] Schiffbautechnische Gesellschaft, 1903. (See also " Mechanical Testing," Batson and Hyde, vol. ii.)
[5] *Proc. Inst. Mech. Engrs.*, July, 1911. (See also " Mechanical Testing," Batson and Hyde, vol. ii.)
[6] Supplied by Messrs. The Lunken Co., London.

The principle used is the differential one, and consists in the observation of the twist between two adjacent points on the shaft by means of two beams of light projected on to a scale from a fixed and movable mirror. The beam thrown on the scale by the fixed mirror is taken as the zero point, whilst that projected by the movable mirror indicates the amount of the torque on the shaft. Both mirrors rotate with the shaft, but even at moderate speeds the

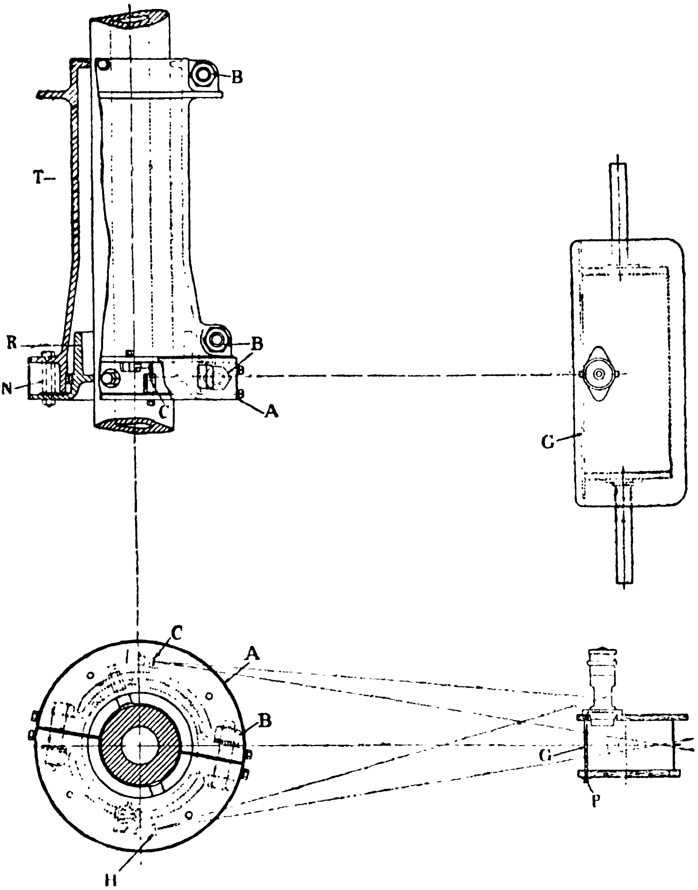


FIG. 154.—The Hopkinson-Thring torsionmeter.

reflections appear as continuous lines of light across the scale, due to the persistence of vision effect.

Referring to Fig. 154, which is a diagrammatic sketch of the apparatus in plan and end elevation, there is a collar R clamped to the shaft to be measured; this collar is provided with a flange projecting at right angles to the shaft, and an extension. A sleeve T, provided with a similar flange and extension at one end, is

clamped at its other end on to the shaft in such a manner that its flange is close to that on the collar R, its extension overlapping that of the collar, on which it is supported to keep it concentric.

When the shaft transmits power the flange on the sleeve T moves relatively to that on the collar R, the movement being equal to that between the two sections of the shaft on which these fittings are clamped. This angular movement is made visible (and is also magnified) by one or more systems of torque mirrors mounted between the two flanges, which reflect a beam of light, projected from a lamp, on to a scale on ground glass.

Each system of torque mirrors consists of a mounting, pivoted

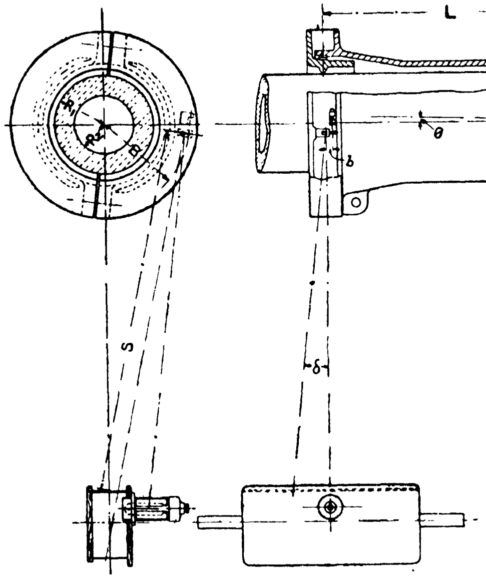


FIG. 155.—Illustrating method of calculating the H.P. in Hopkinson-Thring torsionmeter.

top and bottom on one or other of the flanges in which two mirrors are arranged back to back. This mounting is provided with an arm, the end of which is connected by a flat spring to an adjustable stop on the other flange. Any relative movement of the two flanges will turn the torque mirror and so cause the beam of light to move on the scale, the deflections produced being directly proportional to the torque transmitted. From the rigidity modulus and the shaft speed, the h.p. transmitted can then be calculated.

The h.p. transmitted by the shaft can be calculated direct, from a knowledge of the scale reading x, the modulus of rigidity C, the speed of the shaft N (in r.p.m.), and the constants for the optical system (see Fig. 155).

Thus if S = distance from the scale to the mirror,
B = radius of circle described by the centre line of the arm of the mirror,
L = distance between the clamping planes of the sleeves,
b = length of arm of the mirror,
D = diameter of the shaft,
(all lengths being expressed in inches),

then
$$\text{H.P.} = \frac{\text{scale reading } x}{k} \times \text{N}$$

where k is the horse-power constant, or scale reading given by a torque T_0, producing 1 h.p. per r.p.m.

The value of k is given by

$$k = \frac{25{,}680{,}000 \times \text{SBL}}{\text{C} \cdot b \cdot \text{D}^4}$$

For a hollow shaft of external and internal diameters D_1 and D_2 inches respectively, the quantity D_4 in the above formula is replaced by $(D_1{}^4 - D_2{}^4)$.

Dr. Geiger's Torsiograph.[1]—This instrument has been designed to give actual records, on paper, of the torsional variations which occur in rotating shafts. It is also very useful as a means of indicating the angular vibrations which occur in fixed machinery parts, aircraft, automobiles, buildings, etc.

It is directly applicable to high speed internal combustion engine shafts, and has a speed application range of from 30 to 4000 r.p.m. The resulting records show the torque variations which occur during each revolution.

From such records the causes of erratic stresses, and irregularities, as well as the occurrence of critical speeds, can be studied.

The principle of the instrument can be followed from Fig. 156 (diagrams A and B).

There is a very light aluminium belt pulley c inside which, and concentric therewith, runs a heavy flywheel a. Both the pulley and flywheel run independently on ball-bearings b, but are connected to each other by means of a carefully calibrated volute spring e. When the instrument is in action, the pulley c is belt-driven from the shaft, the characteristics of which are to be examined. The pulley c takes up and reproduces the torsional vibrations and irregularities of the shaft, while the flywheel a tends to run at uniform speed. It follows, therefore, that there will be angular variations between the pulley c and flywheel a; these are transmitted by means of the levers f and i to a tracing pen which traces a diagram on a continuously-moving paper strip.

[1] Supplied by the Lunken Co., London.

The instrument is made very accurately. There is a stratum of air between the flywheel and the pulley, so designed that it damps out the natural oscillations of the latter. The recording levers are very light, work on centres, and are under compression, so that backlash is avoided. The tracing lever is provided with different alternative fulcrums, so that the torque diagram may be obtained to different scales, as required. The paper band can be moved either by clockwork, or by gearing direct off the pulley *c*.

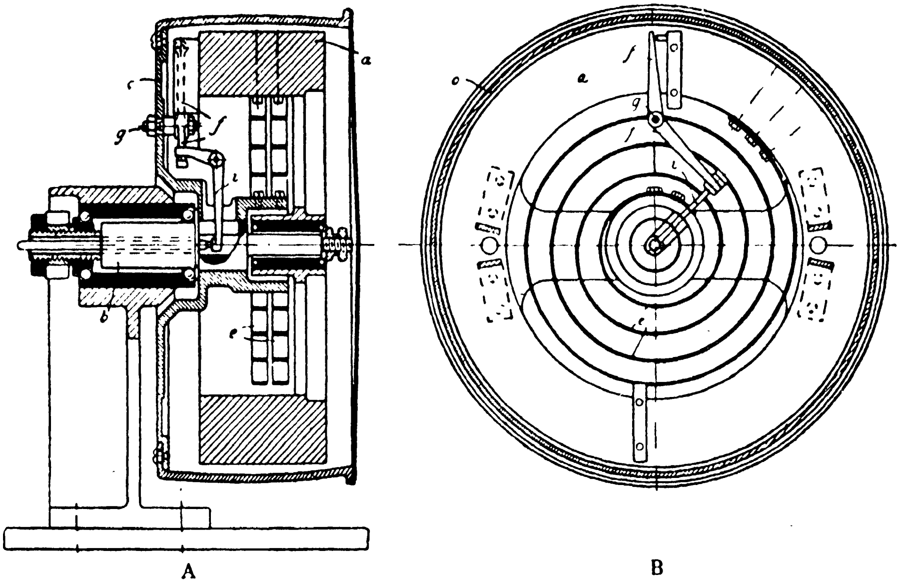


FIG. 156.—The Geiger torsiograph.

Different timing devices are provided with the instrument, as follows :—

1. There is the electrical timing, which marks on the paper the time in seconds and fractions thereof. This is accomplished by the vibrations of a calibrated spring. By this means the speed of the shaft can be ascertained and also the number of oscillations per second or minute. The contact button serves to mark by hand certain places in the diagram. The electrical timing is also used when two or more torsiographs are used on a shaft ; the instruments are then electrically connected, and the timing on all the diagrams occurs simultaneously. It is recommended always to take simultaneous measurements on two or more places of a shaft.

2. The revolution marks can be obtained on the paper at will, showing each turn of the pulley. This is used mainly if the electrical timing cannot conveniently be installed. This device, however,

causes a very slight irregularity in the diagram at each revolution of the pulley.

3. *Timing by hand, which is used on slow-running machinery only.*

4. Dead-centre marks can be obtained by providing the shaft with an electric contact. This is particularly useful for reciprocating engines, pumps, and to investigate the action of propeller blades.

The commercial torsiograph is easy to install and to operate; all that is required is a short length of non-elastic belt. The special belt provided is capable of transmitting vibrations of over 15,000 per minute quite satisfactorily.

Fig. 157 is a reproduction of some typical torque diagrams

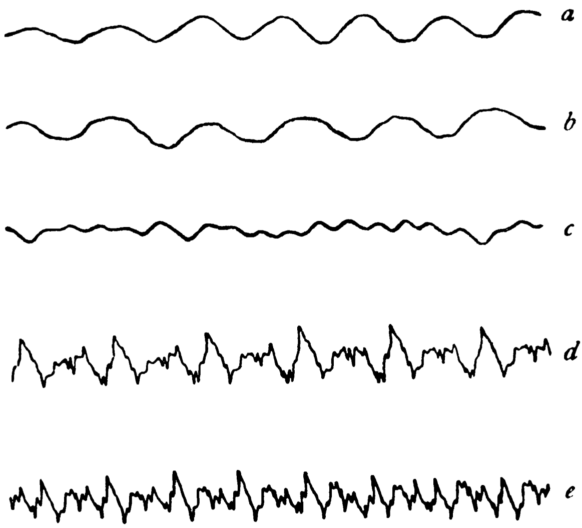


FIG. 157.—Some typical torsiograph records.

obtained with this instrument from a three-cylinder Diesel engine, fitted with a flywheel and dynamo. Diagram (*a*) was taken at the dynamo end of the (250 mm.) shaft; diagram (*b*) between the flywheel and dynamo (350 m.) shaft; diagram (*c*) between the air-compressor and the first engine cylinder; diagrams (*d*) and (*e*) were obtained from the half-time shaft.

The smallest torsional irregularity occurs in the case of (*c*), and the greatest for (*d*) and (*e*), contrary to one's expectations. It is possible from such records to make a careful examination of torsional variations, vibrations, and irregularities, which would otherwise be most difficult either to predict or to compute.

By locking the belt pulley with the device provided, and fixing the instrument to any vibrating body or system, diagrams of the

angular vibrations occurring, giving their frequency and amplitude, are obtained.

Higher Speed Range Torsiograph.—A more recent model Geiger torsiograph [1] (*Fig.* 158) enables vibrational amplitudes over a frequency range of 1000 to 10,000 vibrations per minute to be recorded, using a belt drive and for pulley speeds up to 10,000 r.p.m.; these are much higher frequencies than those obtainable with the normal instrument described previously.

The measuring part of the instrument comprises a light aluminium pulley A of 100 mm. diameter driven by a 2-in. non-elastic belt; owing to its low moment of inertia the pulley A responds quickly to rapid changes in the angular velocity of the driving shaft. A flywheel B is arranged inside A and is elastically connected to the latter by two radial springs C which are not strong enough to transmit fluctuations in the angular velocity of the pulley. Thus, the fly-wheel revolves uniformly whilst the pulley revolves with the varying angular velocity of the driving shaft. The outer ends of C can slide in the slotted heads of the driving pins screwed into the flywheel, whilst the inner ends of these springs are secured in a ring H which is clamped to the hub of the pulley by the clamping nut J. Between the aluminium pulley A and the flywheel B the relative motions correspond to the absolute fluctuations of rotation and are magnified by levers D and E, and transmitted through the hollow spindle of the instrument by rod F to the recording pen. The high-speed torsiograph is arranged for magnifications of 2·55/1, 5·1/1, 10·2/1, and 20·4/1, corresponding to the 3, 6, 12 and 24 magnifications of the normal instrument.

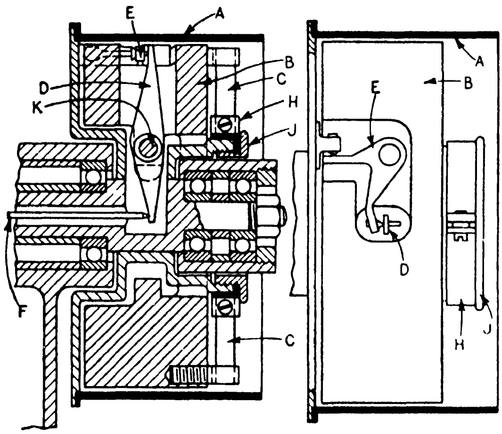


FIG. 158.—High speed range torsiograph.

The recording and timing apparatus is the same as that of the normal instrument, the records being made in ink on a paper ribbon driven either by clockwork or by direct-gearing from the aluminium pulley.

Two timing pens are provided. One is for drawing a damped vibration curve of carefully calibrated frequency which can be either 3000 or 1500 vibrations per minute according to the require-

[1] *Autom. Engr.*, August, 1939.

ments of the particular test in hand. The other is for marking consecutive revolutions of the driving shaft when used in conjunction with a make-and-break device, or for marking ¼ and ½ second timing intervals when used in conjunction with a precision contact clock which can be obtained from the makers of the instrument. *Hardened steel needle points are used for transmitting the movements of one lever to another with the object of minimising friction. The transmitting points are kept in contact by a small* spiral spring at the recording end of the transmission rod F (Fig. 158) which reacts against the flywheel control springs C. This spring equipment is carefully calibrated so that there is no tendency for the transmitting points to separate under the influence of inertia forces, whilst the strengths of the two sets of springs are matched so that the recording pen takes up a mean position on the paper ribbon when no vibration is being transmitted.

Small adjustments in the position of the recording pen relative

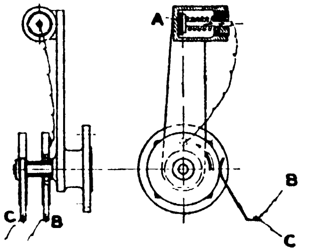


FIG. 159.—Schematic drawing of carbon-disc type torsiograph. A = carbon. B = Electrical connection. C = Sliding contact for earth connection.

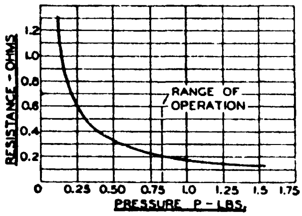


FIG. 160.—Showing relation between carbon-disc pressure and electric resistance.

to the paper ribbon can be made either by the adjusting screw at the recording end of transmitting rod F, or by unscrewing the clamping nut J and rotating the ring H until the recording pen takes up the desired position, when nut J should be securely tightened.

Carbon-Disc Torsiograph.—A more recent design of torsiograph of German origin, due to Dr. Ing. K. Schwaiger and Dipl. Ing. K. Boger, employs the principle of variation of electrical resistance between a carbon disc and metal support. It is claimed to be superior to previous torsiographs based upon different principles and is not subject to detrimental vibration effects.

The principle of the torsiograph is illustrated in Fig. 159.[1] A metallic plunger is carried on an arm secured to the forward end of the crankshaft. The plunger is of mushroom type, its stem passes through a bushing of insulating material, and its head is pressed by a coil spring against a carbon disc, which latter in turn

[1] Automotive Industries, September 24th, 1938.

is pressed against the bottom of the cylinder in which the plunger is located. A current from a constant-pressure source is sent through the junction between the plunger and the cylinder, the instrument being provided with two contact discs through which the current enters and leaves.

When the forward end of the crankshaft is vibrating torsionally, the pressure between the carbon disc and its supports varies accordingly, owing to the forces of acceleration and deceleration impressed upon the plunger. An alternating current is then superimposed on the continuous current which normally flows from the carbon disc to the metal support. If the resistance of the carbon disc changed in direct proportion to the pressure, the value of the alternating current would be directly proportional to the acceleration. Actually there is an hyperbolic relation between the pressure and the electrical resistance, as shown in Fig. 160. It is possible, however, to so adjust the instrument that operation is entirely within one of the comparatively straight end portions of the curve, by varying the force of the plunger spring. The spring, of course, must be sufficiently strong so that the carbon disc will not separate from the metal base even when the acceleration is a maximum. Brushes on the sliding rings did not give satisfactory results, because of changes in the contact resistance due to vibration, and mercury contacts are used instead. The alternating current produced is suitably magnified and then sent through a cathode-ray oscillograph.

In regard to the results obtained with the instrument, two lines are being traced on the record, one being a curve of torsional vibration, whose ordinates are measures of the amplitude of that vibration, the other containing division points equal to one crankshaft revolution. From the number of waves or ripples in the vibration curve between crankshaft revolution marks, the number of the harmonic causing the vibration is at once apparent. A time mark is also recorded, and from that the engine speed and the natural frequency of the system can be determined.

The R.A.E. Torsiograph.—An interesting mirror torsiograph of the polar curve type has been developed by the staff of the Royal Aircraft Establishment, Farnborough, for the purpose of obtaining actual photographic records of torsional vibrations of aircraft engine crankshafts, or crankshaft airscrew combinations.[1]

The apparatus consists of (1) a very stiff (in torsion) actuating tube. (2) A differential nut-locking device to anchor one end of the actuating tube to the crank end of the hollow airscrew shaft. (3) The torsiograph instrument carrying two mirrors, one of which is tilted by the relative angular movement between the actuating tube and airscrew end of the shaft. (4) A camera box and source

[1] *Torsional Vibrations of Crankshafts: The R.A.E. Mk. III Torsiograph*, Aeron. Research Comm., R. and M. No. 1248.

of light to enable records to be made on a photographic plate; these records are lines traced out by the spots of light reflected from the mirrors during the rotation of the crankshaft; and (5) An exposure timing shutter of the gravity type.

Referring to Figs. 161 and 162, the instrument consists of a flange member (1) which is attached to the airscrew shaft by means of an adapter and forms the body of the instrument. Inside the member (1) is fitted a ball-race (13) which carries and centralizes the free end of the actuating tube (12). Another flange member (2) is mounted on a ball trunnion and is arranged to be adjustable for tilt by means of the three set screws (16). This member (2) carries a fixed mirror (9) which serves to provide the base circle on the torsiograms. A small stirrup (6) positioned within the main

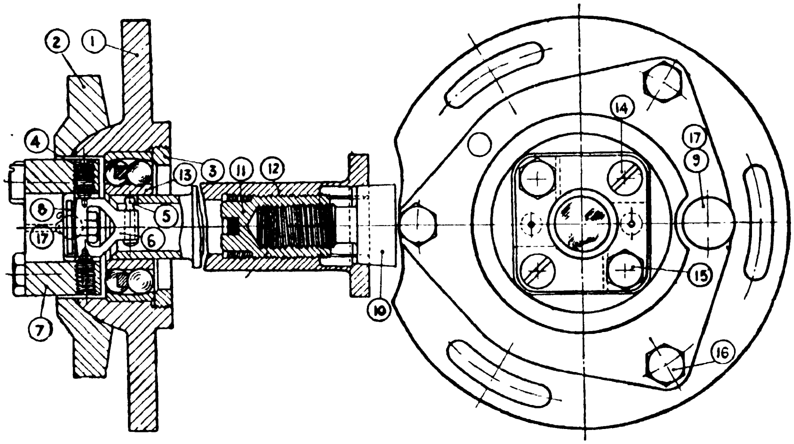


FIG. 161.—Illustrating principle of the R.A.E. torsiograph.

member (1) on conical bearings (4) supports the moving mirror (8) which is locked by a 6 B.A. nut (17).

The bracket carrying the stirrup is adjustable about one axis by the set screws (15) to provide any zero setting required for the beam of light reflected by the mirror. The fixed end of the actuating tube (12) is locked within the shaft at the crank end by the differential nut (11). This locking is achieved by means of a long screw-driver when assembling the apparatus. The free end of the actuating tube (12) which projects through the self-aligning ball-race (13) engages the end of the pin (5) on the stirrup (6) by means of a small tongue which is sufficiently stiff to give a dead-beat motion to the mirror. Fig. 162 shows the instrument fitted to a crankshaft. The engagement is such that any angular movement of the actuating tube relative to the body of the instrument causes the central mirror (8) to tilt about the stirrup axis (4), thus traversing the reflected beam of light across the ground-glass screen or the photographic

plate in the camera box. The focal length of the mirrors is 20 inches, thus giving good magnification, and the source of light is a "Pointolite" lamp. A 500 C.P. lamp consuming 5 amperes at 200 volts gives satisfactory photographs.

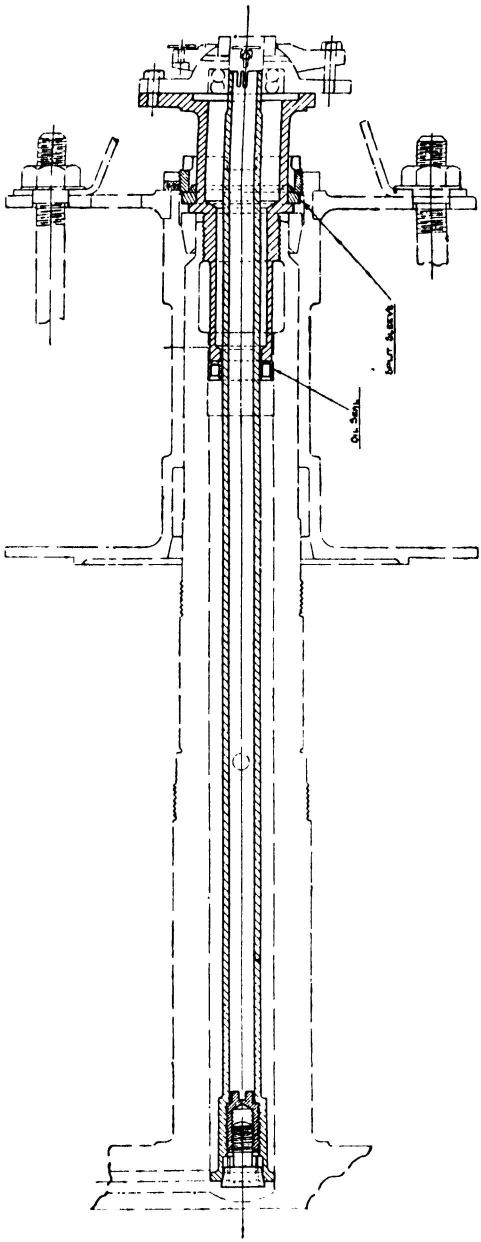


FIG. 162.—The torsiograph fitted to an engine crankshaft.

It will now be understood that any torsional movement taking place between the locked end of the actuating tube and the airscrew hub during rotation will cause a radial deflection of the light beam, the path of which may be followed by visual observation. The fixed mirror at the same time traces out a reference or base circle which serves to show any bending of the shaft or motion other than pure torsion which is superimposed upon the record given by the tilting mirror.

In order to obtain torsiographs which will permit of accurate analysis it is necessary to adjust the exposure of the photographic plate to cover only one or at most two revolutions of the engine at the speed concerned. This is achieved by a falling shutter fitted in front of the source of light which uncovers for the requisite period. The height of fall is adjusted to suit the speed of the engine so that the correct exposure is obtained.

The procedure in obtaining torsiographs is simple, and the main points to observe are those relating to the correct adjustment of the mirrors. When revolving the airscrew slowly to different positions, the spot of light reflected on to the screen by the pivoted mirror should describe a circle of about 2 inches diameter, which represents the setting when no torque is applied. The light spot from the other mirror (9) must be adjusted to give a base circle located a little within the boundary of the screen or plate, and must be positioned radially to that of the pivoted mirror (8). The adjusting screws provided permit these settings to be made with accuracy. The zero positions are marked on the screen for four positions of the airscrew at 90° and recorded for use in the subsequent analysis of the torsiographs.

The instrument now being set, diagrams may be observed on the ground-glass screen under running conditions, with the exposure timing shutter fully open. The engine may be run at various speeds and any tendency towards resonance noted from the shape and magnitude of the peaks of the diagram. The radial distance from the zero position to the diagram is a measure of the torque of the engine at the crank angle considered. Any torsional vibration of the shaft will be evident as irregularities of the inner circle relative to the base circle, and peaks will be produced corresponding to various orders of torsional vibration that may be sufficiently pronounced.

The reader is referred to the original Report mentioned in the footnote on page 207 for further examples of torsiographs and their interpretations.

The Moullin Torsionmeter.—A novel form of torsion measuring apparatus based upon an electrical method of measuring the angle of twist is that designed by E. B. Moullin[1] for both marine and high speed internal combustion engines.

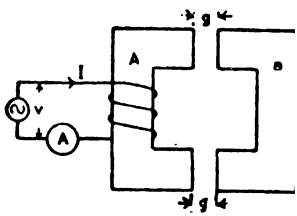


FIG. 163.

Referring to Fig. 163, which illustrates the principle of this instrument, the method employed depends upon the variation of self-inductance of a coil mounted on the torsionmeter. There is an electromagnet consisting of two similar laminated iron cores of rectangular cross-section : a coil of insulated wire is wound round one half of the core, and the two halves are separated by an air-gap *g*. If an alternating voltage V be applied between the terminals of the winding, the current I passing, measured by the ammeter A, depends to a negligible extent on the ohmic resistance of the winding but almost entirely on the

[1] "A Distant Indicating Torsionmeter," E. B. Moullin, *World Power*, Oct. 1925.

number of turns in the coil, the cross-section of the core, and the length of the air-gap g. If the air-gap g is increased the current

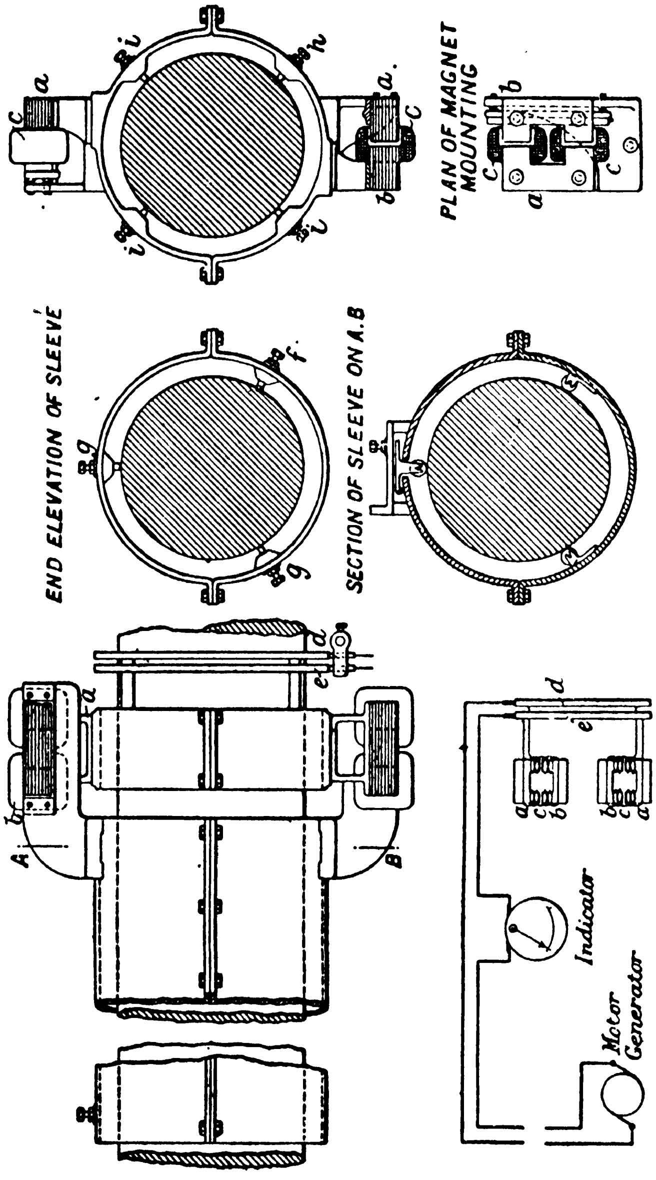


FIG. 164.—Illustrating the principle and constructional details of the Moullin torsionmeter.

will increase, and if the electromagnet, generally called a choking coil, is suitably designed the rate of increase of current with gap is constant over a large range.

If a suitable type of ammeter is selected the gap movement can be magnified—as shown by the scale reading—at least 100 times.

Further, since the system is electrical we may have the two halves of the choking coil revolving, if need be, in one place and the ammeter in another; thus the instrument can be made a distant reading one.

Referring to Fig. 164, the half of the choking coil marked A in Fig. 163 is attached to the tubular pointer B, whilst the part marked *a* in Fig. 163 is attached to the shaft at the part marked *a* in Fig. 164.

The current is led in and out of the winding by brushes pressing on two insulating slip-rings revolving with the shaft. The arrangement employed is shown diagrammatically in Fig. 164, at the bottom left-hand corner; this diagram shows also the choking coils mounted on the torsionmeter. As the shaft twists the air-gap is opened (or closed if preferred) and the amount of twist indicated on the ammeter.

This instrument has an additional advantage in being independent of changes of resistance or voltage supply.

If the alternating current ammeter is replaced by a Duddell oscillograph it is possible to obtain photographic records of torque pulsations. A typical example of such a torque diagram is given in Fig. 167, which shows the torque values for a six-cylinder four-stroke Diesel engine. Referring to Fig. 164, it may be seen that the torsionmeter is fitted with two choke coils, one at each end of a shaft diameter: only one coil is essential, but there are several advantages in fitting a pair. Firstly, mechanical balance is obtained, and this is important as it equalizes the load on a mounting so difficult to make rigid. Secondly, it is a device which compensates for small parasitic movements in the mounting. The two choke coils are connected in parallel electrically, and on reference to the top right-hand illustration of Fig. 164, it may be seen that if the tube moves slightly along a horizontal diameter of the shaft, then one gap will be closed and the other opened by an equal amount. The two coils are similar and have straight line gap-current characteristics, and consequently the effective inductance of the two coils in series remains unaltered and no change of current is produced: a small vertical or axial movement of the tube produces only a second order change of inductance.

Fig. 165 shows the arrangement of a torsionmeter made for testing aircraft engines running: it is convenient to load these engines with their own propeller, which produces the necessary cooling draught, and then the load cannot be measured except by a torsionmeter. This figure shows the constructional details of the high speed mounting, and also the method of housing the choking coils.

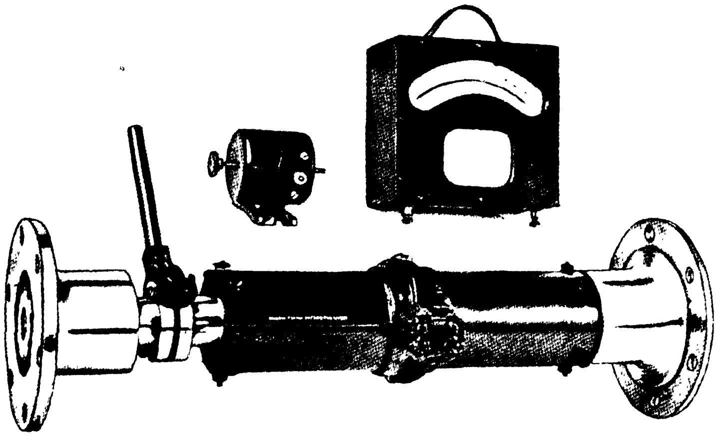


FIG. 166. The complete Moullin torsionmeter.

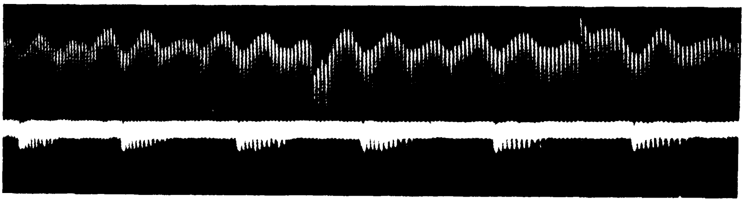


FIG. 167.—Torsiograph record from a six-cylinder four-stroke Diesel engine.

[To face page 212.

Fig. 166 shows the instrument mounted on its shaft and also the indicating dial and alternator. The indicating dial *can be calibrated* by twisting the shaft statically, or the shaft may be coupled to a Froude brake, and the latter's readings compared with those of the torsionmeter.

Measuring Aircraft Engine Power in Flight.—Although much important information can be obtained from laboratory tests of aircraft engines, it is by no means an easy matter to ascertain the power outputs under the varying conditions of altitude, temperature, and humidity met with in flight. Frequently, also, the behaviour of the engine depends to some extent upon the aerodynamic characteristics of the airscrew under flight conditions.

For this reason several attempts have been made to measure the actual engine horse-power developed whilst the machine is in flight.

The "Farnboro" electric indicator, as we have already stated, has been used for this purpose, but obviously it does not provide so convenient a method as the direct torque reading one.

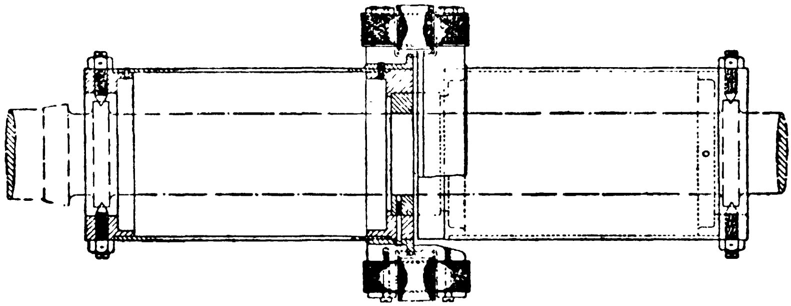


FIG. 165.—Showing method of fitting Moullin torsionmeter to shaft.

Official tests made in this country have shown that it is possible to devise apparatus to measure the average engine torques under flying conditions, and with a knowledge of the corresponding engine speeds the power developed by the engine has been estimated.

The following are descriptions of the D.V.L.[1] hydraulic torque meter developed in Germany several years ago, and used more recently by the American aircraft authorities.[2]

The Bendemann dynamometer unit in question was a special airscrew hub in which was incorporated a series of pistons and cylinders interposed between the airscrew and the engine crankshaft. Both the *torque* and *thrust forces* could be balanced by fluid pressures in a closed hydraulic system recorded by instruments in the observer's cockpit.

[1] Deutsche Versuchsanstalt für Luftfahrt.

[2] "Measurement of Engine Power in Flight with a Hub Dynamometer," N.A.C.A. Report No. 252.

The tests made with this hub dynamometer showed that it was suitable for measurement of the power in flight and for the determination of the torque and power coefficients of the propeller.

The dynamometer hub replaces the conventional airscrew hub and allows the airscrew to be mounted in its original position relative to the engine. The dynamometer mechanism is placed just ahead of the airscrew, as shown in sectional views, Fig. 168. It consists essentially of: (1) a steel sleeve adapter keyed to the engine crankshaft; (2) a cast-steel airscrew sleeve which is a loose fit on the adapter; (3) a cast-steel cylinder block which is keyed to the adapter and contains the torque and thrust cylinders and drilled passages for transmitting working fluid to the cylinders; (4) a bronze spider and revolving hub containing drilled passages which register with the passages in the cylinder block to which it is bolted; and (5) a stationary spindle which contains drilled passages connecting with annular grooves in the revolving hub at one end and with tubes connecting the dynamometer hub with the recording instruments at the other end.

The airscrew sleeve is a running fit on the crankshaft adapter and is free to oscillate a few degrees and to slide axially a distance of $\frac{1}{8}$ inch. Its rotational motion is limited by metal stops on the cylinder block and its axial travel is restricted by the rear shoulder on the adapter and the cylinder block. The stops mentioned serve to drive the airscrew in case of loss of liquid in the hydraulic system. Steel piston-rods having hemispherical ends which rest in steel sockets at either end are interposed between the pistons in the cylinder block and the propeller sleeve and are of such length as to hold the airscrew sleeve free of the stop lugs when the pistons are in working position.

The working fluid is introduced under all the pistons, and the fluid pressures generated in the cylinders by the torque and thrust forces are recorded.

Fig. 169 is a diagrammatic sketch of the complete dynamometer apparatus and shows the operation of the hydraulic system. A high-pressure supply of fluid is used to compensate for any leakage in the systems. Since each set of cylinders has connecting passages, admission of supply fluid is controlled by only one automatic valve on each set of three measuring pistons.

This valve is only opened when leakage causes the piston to approach the cylinder head and closes as soon as the fluid is replaced. This regulating valve is the outstanding feature of the dynamometer, and the regulation is adjusted to such a nicety that the motion of the piston is barely perceptible, and the indicator records do not show when the new fluid is admitted.

The rotating hub and the stationary tubing of the thrust, torque, and supply systems are connected as shown in Fig. 169. The lines

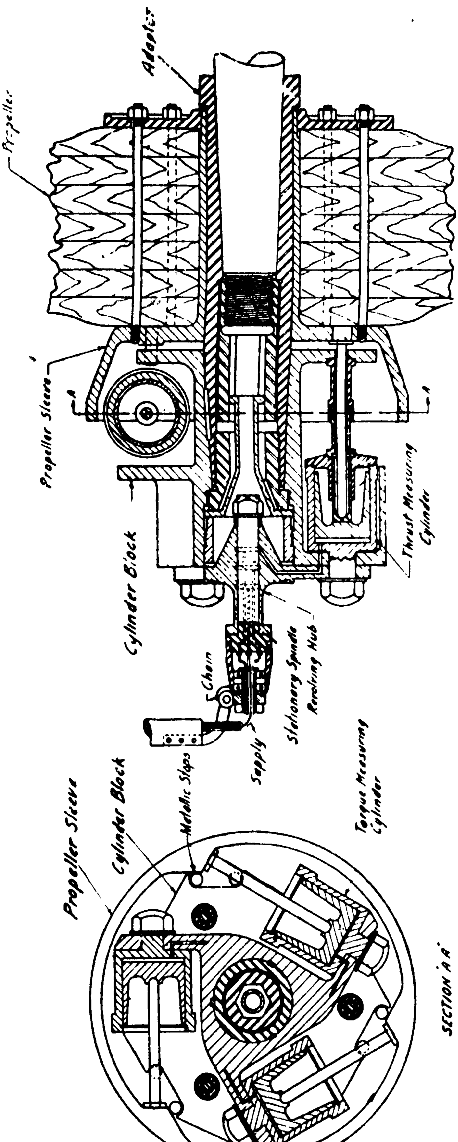


FIG. 168.—The Bendemann dynamometer hub.

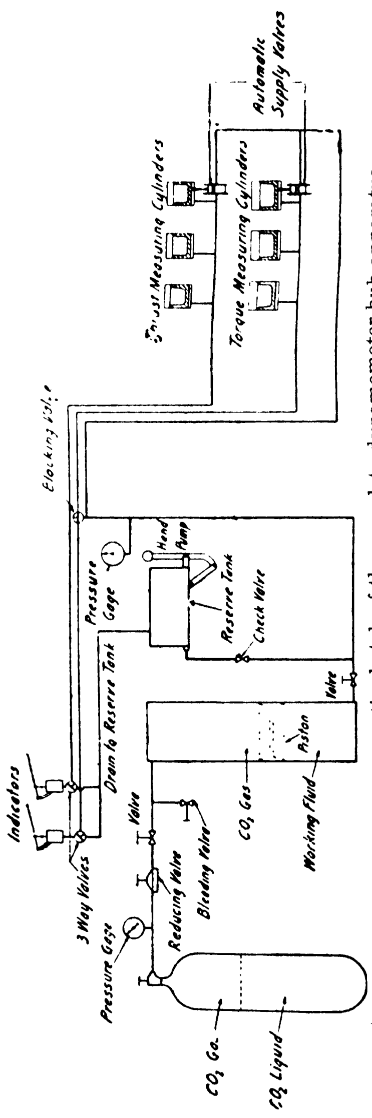


FIG. 169.—Diagrammatic sketch of the complete dynamometer hub apparatus.

are connected to a stationary spindle whose radial holes register with annular grooves in the revolving hub. It is necessary to carry the tubing from the stationary spindle over the airscrew to the controlling and recording apparatus in the cockpit. The torque and thrust lines lead to their respective recording instruments, each of which consists of a spring-loaded piston actuating an indicator arm carrying a brass stylus which traces a line on metallic-faced paper. A drum, revolved at constant speed by a clockwork, carries this paper. A third traces a reference line.

The supply line is served by a tank containing the working fluid. Pressure in the tank is maintained by gas acting on a piston, the latter serving to separate the gas from the working fluid. Liquid CO_2 was used to supply the gas pressure, which was maintained at about 425 lb. sq. in.; the pressures in the torque and thrust systems never exceeded 275 lb. sq. in. The hub dynamometer was calibrated in the laboratory, before fitting to an aeroplane, against applied torque.

A fan brake was substituted for the airscrew and the hub was set up so that it would be driven by an electric dynamometer. In addition, static load calibrations of torque were made by holding the shaft rigid and applying known values of torque to the airscrew sleeve.

For further information on the subject of dynamometer hubs the reader is referred to the footnotes below.[1] [2]

Measurement of Engine Losses.—It has been stated that the total engine loss of power is the sum of the frictional (or mechanical) losses and the pumping losses. A knowledge of the engine losses enables the mechanical efficiency of the engine to be computed from the B.H.P., or from the I.H.P. An analysis of the total into its component losses provides valuable information concerning the design of the components, and reveals, frequently, sources of power loss unsuspected.

The total losses may be determined in two different ways, namely :—

1. By measuring both the I.H.P. and the B.H.P. and taking the difference, or
2. By measuring the losses, directly or indirectly.

In the present stage of development of the high speed indicator, the former method is employed only in research laboratories. The latter method is more general, although, as we shall see, for accurate results, the use of an indicator is essential. The usual method of determining the engine losses is to motor the engine around whilst it is warm, and to measure the power required to turn the engine

[1] "Dynamometry with Particular Reference to the Measurement of Engine Power in Flight," N. S. Muir, *Journ. Roy. Aeron. Soc.*, Feb. 25, 1937.

[2] "The Variation in Engine Power with Altitude Determined from Measurements in Flight with a Hub Dynamometer," W. D. Gove, N.A.C.A. Report No. 295.

at various speeds. The engine is run under its own power for a sufficient period of time to enable the metal, oil, and water to attain their ordinary working temperatures. The swinging-field, electric type dynamometer is particularly well adapted to these tests. The ignition and water circulation are then switched off, the electrical connections to the dynamometer changed over by means of a suitable switch, to convert it into an electric motor, and the engine is then motored at the speed at which it had previously been running, under its own power. The power required to motor it at this speed is measured on the torque-arm.

The power thus measured does not represent the true engine losses for the following reasons:—

1. The engine is cooling down, due to the cold air being drawn into the cylinders; the maintenance of the circulating water at its normal temperature to some extent counteracts this. The engine oil, on the above account, becomes cooler and more viscous, so that an increase in the piston friction, for a certain period after switching off and motoring, is to be anticipated.

2. The pressures on the bearings and the thrust on the cylinder walls are different when the engine is motored; in several cases they are reversed in direction. The absence of the high explosive and expansion pressures reduces the loads on the connecting rod and main bearings.

3. The power required to draw in the air and to expel it during the exhaust will be different to that required under running conditions. The exhaust back pressure has been shown to be greater during motoring tests than its normal value, due to the fact that there is no high pressure at the time of discharge to supply the kinetic energy necessary to propel the exhaust products down the exhaust pipe.

4. The air drawn in has a lower temperature during the induction stroke, due to the absence of the heated exhaust gases. A rather greater quantity of air will, therefore, be induced. This air is compressed and then expanded; it loses heat to the cylinder walls during the former process, and it does not regain this during the following expansion process. The result of this is that the expansion line on the indicator diagram lies below the compression line; the area included between these two lines represents the power expended in compression and expansion. This power must be deducted from the total measured power in the computation of engine losses. It usually amounts to 0·30 to 0·50 h.p. at 1000 r.p.m. for a 20-h.p. four-cyllinder engine. The power required to induce and expel the air, in item (3), can be ascertained from the area of the suction-exhaust loop of the indicator diagram. The area of this loop should be measured (*a*) when the engine is firing, and (*b*) when it is motored at the same speed. Any differences found should

be allowed for. From these facts it follows that the engine losses obtained by the motoring test must be corrected for the four enumerated items. If we denote the difference between the mechanical losses when firing, and when motoring, by F h.p., the difference in the induction-exhaust powers by S h.p., and the power absorbed in compressing and expanding the air by C h.p., then the engine losses E will be given by

$$E = (M \pm F - S - C) \text{ h.p.}$$

where M is the motoring power.

Ricardo states that there is actually a decrease in the piston friction following the switching off of the ignition and during the motoring test. At first the friction is rather high, but drops gradually until, after a period of about fifteen minutes, it has fallen to from 10 to 20 per cent. of its normal value. This may be due to the gradual change from the used oil to fresh oil on the cylinder walls, and without temperature change or to the reduced thrust on the cylinder walls.

He considers, further, that the reduced piston friction (which accounts for more than half the total mechanical losses) balances, on the average, the increased pumping losses, when motoring ; this is stated to be true over a wide range of speed. On this assumption the above expression would be modified as follows :—

$$E = M + F - (S + C)$$

and $F = S + C$

so that $E = M.$

That is, the motoring losses would equal the engine losses. It is stated that the results of alternative methods and cross-checks confirms the conclusion that the true engine losses agree very closely with those measured by the motoring method.

The results of some later tests made at the Air Ministry Laboratory [1] on an aircraft engine showed that I.H.P. given by the addition of the brake and motoring horse-powers was too low at high speeds and too high at light loads. In these tests an optical indicator was used to obtain the I.H.P., and an electric dynamometer to measure the B.H.P. and the motoring h.p.

It was found that the pumping losses when motoring were usually greater than when under power ; this tended to make the sum of the B.H.P. and motoring h.p. greater than the I.H.P. At high loads, however, the increase in sliding friction counteracts this effect ; in the tests in question the two balanced at 70 lb. sq. in. M.E.P. Above this value the sum of the brake and motoring horse-powers was less than the I.H.P. The error at full load was considered to be due to friction increase.

[1] "The Mechanical Efficiency of an Internal Combustion Engine," H. Moss and W. J. Stern, *Autom. Engr.*, March, 1925.

In this respect, the results obtained with the optical indicator were checked by means of the "Farnboro" (R.A.E.) electric indicator. Fig. 170 shows how the sliding friction increases with the indicated M.E.P., i.e. with the load.

Analysis of Engine Losses.—It has been stated that the total engine losses comprise the mechanical and the pumping losses.

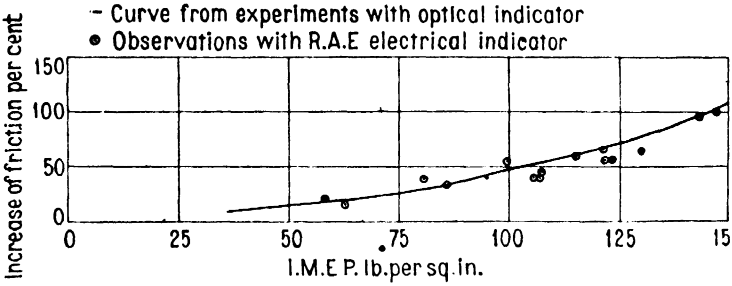


FIG. 170.—Sliding friction—M.E.P. curve.

The results of various investigations have established that these losses—and therefore the total power loss—increase with the engine speed, but at a greater rate than the linear one. The results of motoring tests [1] made upon a single-cylinder petrol engine of 7·25 in. bore and 8·5 in. stroke running at 1400 r.p.m., with a piston speed of 1980 ft. per min. and gas velocity through the valves of 130 ft. per sec., were as follows :—

TABLE XIII

Analysis of Engine Losses

Frictional Losses in Terms of lb. per sq. in. of Piston Area	Valve
Bearings, etc.	1·8
Piston friction	7·2
Fluid pumping loss	3·4
Total	12·4
Brake M.E.P.	118·0
Mechanical efficiency	90·6 %

It will be observed that the piston friction accounts for rather more than 50 per cent. of the total engine losses, the pumping losses being approximately 27 per cent. of the total.

The manner in which these different losses vary with the engine speed is shown in Fig. 171,[2] for a speed range of 1500 to 4500 r.p.m. It will be observed that, with the exception of the bearings and

[1] "The High Speed Internal Combustion Engine," H. R. Ricardo, *Autom. Engr.*, May-July, 1925.

[2] "Handbook of Aeronautics," vol. ii (Pitman, Ltd., London).

auxiliary losses—which show practically a linear rate of increase—the other losses increase rapidly with the speed.

Analysis of Frictional Losses.—The manner in which the various contributory factors affect the frictional losses has been investigated by Lichty and Carson [1] in the case of a four-cylinder motor vehicle engine of $4\frac{1}{4}$ in. bore and $5\frac{3}{4}$ in. stroke. The engine was fitted with special piston rings which during previous tests had shown a reduction of 10 per cent. in the ring friction over the standard ones.

The frictional losses, at different speeds, were determined from

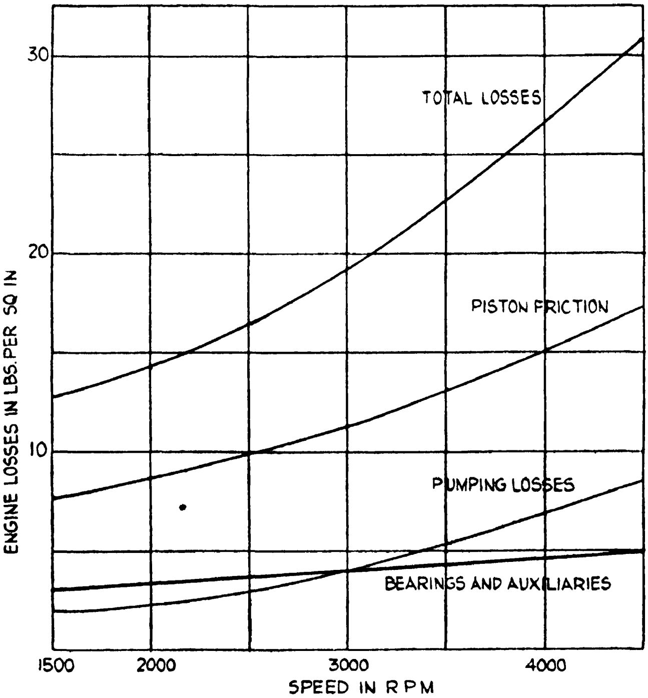


FIG. 171.—Engine losses at various speeds.

a large number of motoring test runs made with the engine complete and then with various components removed in turn.

Fig. 172 shows the results obtained with the cooling water in the jackets at 60° F. and an oil temperature of 150° F. Tests were also made with higher water temperatures, and it was stated that the pumping losses were not affected by the jacket temperature, and that this holds approximately for all other losses except piston and ring friction; the latter decrease materially with an increase in jacket temperature.

[1] *Jour. Soc. Autom. Engrs.*, 1933-34.

At a jacket temperature of 60° the friction with all the rings removed was practically the same as with all the rings in place. When the oil ring was removed the friction was greater than when it was in place, and this is ascribed to the uncontrolled film of oil all around the piston, which greatly increases the shearing effect.

At the low jacket temperature the friction loss due to the reciprocating parts was 7·9 h.p. at rated speed, and at the high jacket temperature, 180°, it dropped to 4 h.p., a decrease of 49·3 per cent. This, the authors believe, is a definite indication that

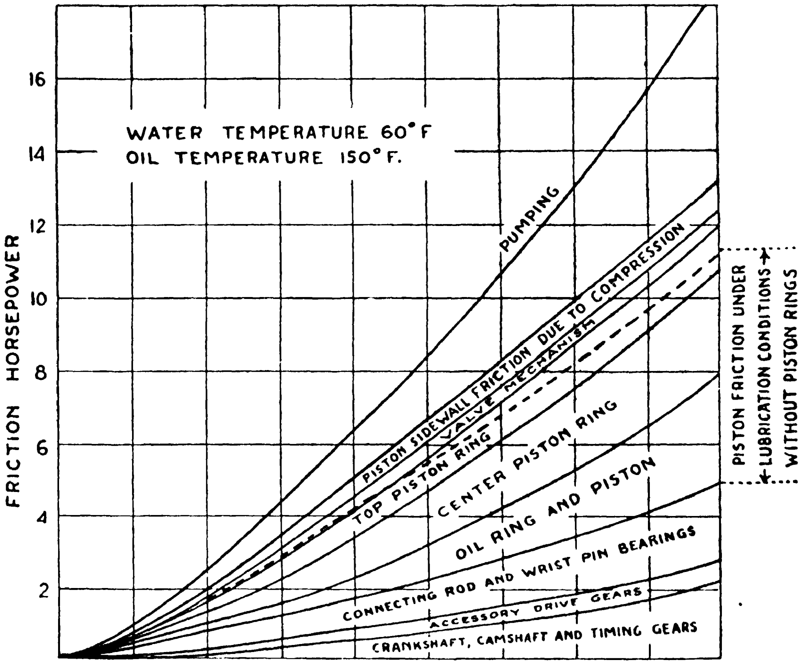


FIG. 172.—Analysis of motor vehicle engine losses at different speeds.

cylinder-block temperature controls the temperature of the oil film on the cylinder wall, and that the shearing of the oil film by the reciprocating parts in all probability makes up the major portion of the friction losses due to the reciprocating parts when operating at low engine temperatures. The loss due to the reciprocating parts was about 59·8 per cent. of the total mechanical friction loss at the low jacket temperature, and about 43·5 per cent. of the entire frictional loss of the engine. At the high jacket temperature these ratios were 43·5 per cent. and 28·6 per cent. respectively.

The conclusions drawn from the analysis are that the friction of the reciprocating parts has a preponderating effect in determining

the overall efficiency of the engine and that the compression rings are responsible for most of the friction of the piston and ring assembly.

Diesel or C.I. Engine Losses.—At the time of writing little reliable information is available on the subject of high speed Diesel engine losses, but the results of some preliminary work mentioned in the 1930-31 Report of the Aeronautical Research Committee refer to some data obtained from experimental aircraft Diesel engines.

It states that considerable uncertainty exists as to the true value of the mechanical efficiency of these engines. The method adopted with petrol engines—that of adding the total motoring loss to the brake output to obtain the indicated horse-power—is known by numerous cross checks to give a reasonably accurate determination, but when this method is applied to compression-ignition engines it appears, in many cases, to give a gross over-estimate of the mechanical and pumping losses. Experiments were put in hand with the object of making as accurate a determination as possible of the mechanical efficiency, and although it was not found possible to evaluate the actual running losses within as close limits as had been hoped, the experiments showed that the losses registered by motoring a compression-ignition engine exceed considerably the actual losses when the engine is running under its own power. The principal source of divergence as between motoring and running conditions was found to be in *the loss of heat towards the latter part of the compression and the early part of the expansion strokes when motoring*. The use of a reliable high speed indicator therefore appears to be the rational method of determining the engine losses.

Alternative Methods.—An alternative method of measuring the engine losses, in the case of a multi-cylindered engine, is to run the engine at full throttle under load, after a preliminary warming-up period, until everything is warmed up normally, and conditions have settled down. The ignition of one cylinder is then short-circuited, and the brake horse-power of the engine is then measured.

The difference between this value and the normal value for all of the cylinders firing is the indicated power of the cylinder not firing.

By repeating the process for each cylinder in turn the indicated power of the complete engine can be ascertained. The engine losses measured in this way are nearly always lower than the actual losses, however.

An alternative method was employed by Ricardo, in the case of the variable compression engine, which was fitted with a heavy flywheel and a built-up crankshaft mounted on ball-bearings. The crankpin was removed (thus disconnecting the pistons and connecting rods) and the flywheel was motored up to, say, 2000 r.p.m. The dynamometer was switched off, and the deceleration noted. Next, the crankpin was replaced and the engine run under full load conditions at the same speed, namely, 2000 r.p.m. It was then

switched off and the deceleration again noted. From the difference in the decelerations the total mechanical losses could be computed, over the speed range from 2000 r.p.m. to zero.

General Considerations.—It is necessary, in the case of motoring tests, to measure the power as quickly as possible after the engine has been switched off. The value of the torque obtained during the first minute or so agrees fairly closely with that for the true mechanical and pumping losses obtained by more elaborate methods.

It is necessary to "switch" off the water injector or pump at the same time as the ignition, in order to keep the cylinder walls under the same temperature conditions during the motoring tests.

Measurement of Component Losses.—Subject to the considerations dealt with in the preceding paragraphs on the accuracy of the motoring test for measuring engine losses, it is possible to make an analysis of the total engine losses and thus to indicate the power losses due to the different components.

Such an analysis is invaluable from the engine designer's viewpoint, as it enables him to concentrate upon those items showing appreciable power losses.

The method originally employed by the late Professor W. Watson was to measure the total engine losses by the torque-arm method, using an electric motor for the purpose. The motor could readily be converted into a dynamo for absorbing the engine's output.

The motoring tests were made immediately after the engine had been run under its own power, with the jacket water at its normal working temperature ; by means of a supplementary heating system it was possible to keep the jacket water at its normal temperature during the tests for competent power losses.

Motoring tests were made at various speeds so as to give sufficient data for a graph of engine losses to be plotted on a speed base.

By stripping one part of the engine at a time, so as to place each particular component out of action, and then repeating the motoring test it was possible to ascertain the power absorbed at different speeds by each component. For example, if after the complete engine loss motoring test the magneto drive is uncoupled, and another motoring test made, the differences in the power readings at each speed can be regarded as representing the power absorbed in driving the magneto.

The valve tappets can be removed to ascertain the power required to drive the valve gear. In turn the pistons and connecting rods, oil and water pumps, etc., can be removed until only the crankshaft remains.

As we have stated previously, it is not an easy matter to measure piston friction accurately in this manner, owing to the fact that the conditions as to the temperatures and pressures in the cylinders are quite different in the motoring and power running tests.

Fig. 173 illustrates the results of a number of engine component power loss tests made in the above manner, by Morris Engines Ltd.,[1] on a four-cylinder side-valve engine of normal production type. A Froude electric-dynamometer of the torque-arm type was used for these tests.

The upper graph shows that 9·25 h.p. is absorbed in internal losses at 3000 r.p.m. It will be noticed that the curves shown are complementary, i.e. the figures on which the curve giving the compression pressure and induction pipe losses are based are subtracted from the total loss curve, and so on.

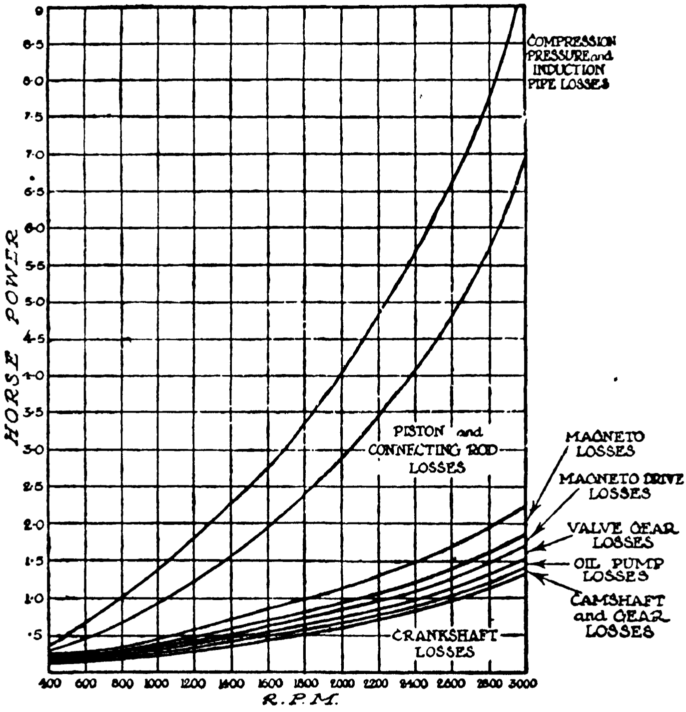


FIG. 173.—Component losses in four-cylinder engine.

For example, the loss due to the causes we have just referred to is, at 3000 r.p.m., the difference between 6·95 and 9·25 h.p.

In order to show more clearly the individual losses, a second series of graphs (Fig. 174) has been plotted, in each case from a common base line. Examination of these last produces some interesting information. Perhaps the most striking point is the power lost in merely rotating the crankshaft in its bearings, no less than 1·25 h.p. at 3000 r.p.m. The greatest loss of all is in the pistons and connecting rods, 4·65 h.p. At the same speed the power

[1] "Engine Power Losses," *The Autocar*, July 16, 1926.

used in rotating the magneto is 0·35 h.p., and in actuating the magneto driving gear 0·15 h.p. The losses in the camshaft and in the valve gear are relatively small.

It is interesting to note that the oil-pump plunger requires 0·1 h.p. at 3000 r.p.m. to operate it.

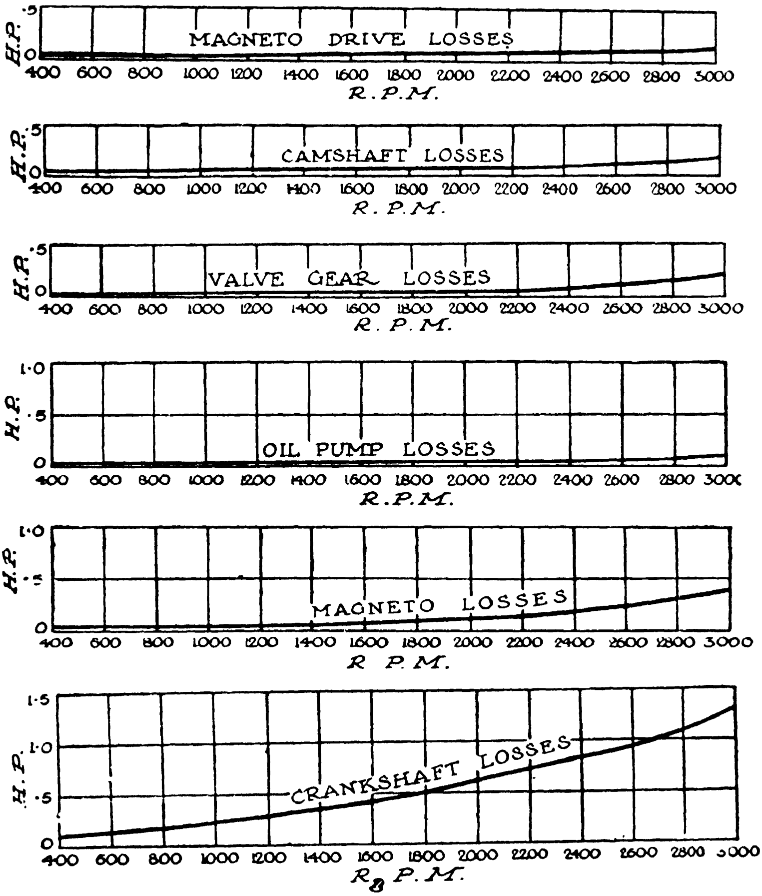


FIG. 174.—Power loss—speed curves for different engine components.

Temperature of Engine Oil.—In connection with tests of the above nature it is important to observe that the temperature of the engine oil should be about the same as the normal working value; this is particularly important in the case of the bearings and oil-pump loss tests.

CHAPTER VII

PRESSURE MEASUREMENTS

A KNOWLEDGE of the pressures occurring in the cylinders of high speed internal combustion engines is essential from the points of view of design, performance, and research work. In addition, it is occasionally necessary, in connection with induction and carburation tests, to be able to measure the pressure variations at different parts of the induction system, whilst silencer and exhaust-operated supercharger tests frequently require pressure diagrams from the exhaust manifold.

In the case of two cycle engines, employing crankcase compression of the charge, the cyclical changes of pressure within the crank chamber have an important bearing upon the engine's performance.

For tests of engine losses, induction pipes, etc., the light spring indicator diagram has proved an invaluable aid to the test engineer and research worker.

Pressure Indicators and Recorders.—The available apparatus for making pressure measurements in engines of the type under consideration may be divided into two principal classes, as follows: (*a*) Single-pressure Indicators; and (*b*) Cyclical-pressure Indicators.

Class (*a*) includes compression, explosion, and mean-pressure measuring devices, which, as a rule, are comparatively simple; whilst Class (*b*) includes the more elaborate apparatus for actually recording the pressures throughout their cycles of changes.

Single-pressure Indicators.—The present types of high speed engine indicator .or cyclical-pressure records are fairly elaborate, and in many cases require laboratory conditions and skilled attention for their correct operation. Most of these instruments are beyond the scope of the ordinary engineer and mechanic, who require a sturdy, robust piece of apparatus capable of standing up to workshop, test-bench, and occasionally, to road conditions. Moreover, the instrument must be capable of ready attachment to the engine, without the use of special fittings or supplementary apparatus.

The engineer usually requires a knowledge of the compression pressures (at different speeds) and explosion pressures which occur in each cylinder of the engine. He can then ascertain whether each cylinder is functioning properly and the effect upon the running of carburation and ignition adjustments.

As a rational means of "tuning" an engine, the single-pressure type indicator is undoubtedly of much utility. It is, of course,

possible to calculate the compression pressure of an engine during the design stage, from a knowledge of its compression ratio, but it is not possible readily to take into account such factors as casting deviations, induction system, unequal distribution, and wire-drawing or throttling effects, premature heating of the charge, and leakages. Moreover, the measurement of the clearance volume is not an easy matter, so that one has to look for a more direct means of ascertaining the actual compression pressures.

The requirements of an efficient single-pressure indicator are that it must be strongly made, capable of ready attachment to the engine, by the simple expedient of fitting into a sparking-plug or compression tap hole, must be unaffected by the temperature of the cylinder, and uninfluenced by the rapidity of cyclical-pressure variations. It should also, for preference, be direct reading with a lb. per square inch scale.

The ordinary Bourdon type of pressure gauge is not suitable for any other but static pressure indications, as, owing to the inertia of its moving parts, it generally gives readings much in excess; it is also apt to wear, develop backlash, and to alter its accuracy.

Some General Considerations.—Of the various types of maximum-pressure indicator described in the following pages, those giving the most accurate results for the maximum cylinder pressures are the electric balanced-pressure diaphragm or disc-type indicator, so constructed as to have a diaphragm or disc of relatively large area and minimum seat width and mass.

Tests made on a number of different types of maximum-pressure indicator [1] show that the values of the maximum pressure given by these instruments vary appreciably for the same pressure conditions. It is the high pressure of short duration that is most difficult to measure.

The types of indicator available may be grouped into two principal classes, viz.: (1) those that make use of the cylinder pressure in recording directly, i.e. all the work of recording being done by the gas in the cylinder; and (2) those that use the cylinder pressure to operate only an auxiliary part of the recording apparatus.

The piston type indicators belong to the former class, whilst the balanced disc type, e.g. the "Farnboro" indicator, comes into the latter class.

In the former class the main sources of errors are the inertia of the moving parts, piston friction, and temperature effect on the load spring.

The electric balanced disc type gives the more accurate results, but an error is introduced due to the disc seating width giving different areas exposed to the gas and balancing pressures. This

[1] "The Measurement of Maximum Cylinder Pressures," C. W. Hicks, N.A.C.A. Report No. 294, 1928.

error can, however, be reduced to relatively small proportions by suitably designing and arranging the disc on its seating. Alternatively, a suitable correction can be applied to its readings.

The Gibson Pressure Indicator.—The disadvantages of the ordinary type of Bourdon pressure gauge, when exposed to rapidly fluctuating changes of pressure, to which we have referred earlier in this chapter, can, to a certain extent, be overcome by introducing a non-return valve between the cylinder and the gauge, so that the pressure is accumulated or "piled up" on the gauge side instead of fluctuating violently if there were no valve. In this manner the maximum pressure can be indicated with little or no oscillation of the gauge needle. Fig. 175 illustrates the Gibson gauge, which employs this method. The non-return valve is shown at B, whilst at C is shown a four-way cock for connecting each of the cylinders of a four-cylinder engine with the gauge in turn.

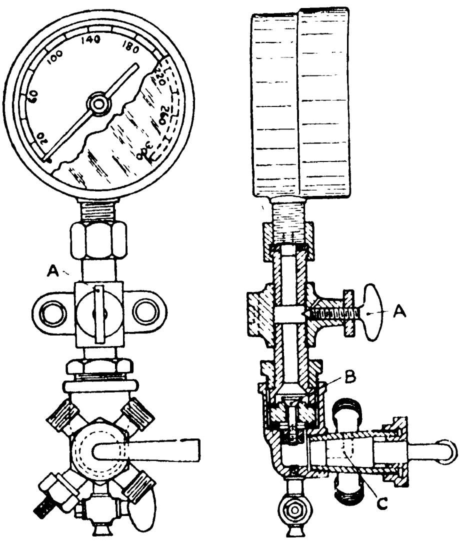


FIG. 175.—The Gibson maximum-pressure gauge.

A drain tap and a pressure-release valve A are provided also.

It is necessary, with instruments of this type, to reduce the lengths of the communicating passages to the cylinders to a minimum in order to avoid any pressure-restriction effects.

The Okill Indicator.—One of the simplest types of single-pressure indicator is the Okill instrument, illustrated in Figs. 176 and 177, the design shown being an improvement on the earlier micrometer-reading one.

Referring to Fig. 176, there is a screwed part S, so arranged that the instrument can replace the sparking plug, or compression tap, the body A being provided with a hexagon for taking a spanner.

The piston B is an accurate working fit in a cylinder formed in A, and is spring-loaded by means of the hollow piston-rod (and oil tube) and spring shown. The pressure, or load of the spring, which is communicated to the piston B by means of the collar shown, can be varied by screwing the milled-headed cap up or down.

In the earlier designs the pressure scale was arranged as a micrometer, the fixed scale being engraved on the outer surface of the barrel C, and the moving scale on the circumference of the outer

sleeve D. In the present instrument the movement of the milled-headed cap is communicated by means of a train of three pinions to a simple counter, which is arranged to indicate the pressure in lb. per square inch, corresponding to the load on the piston B, divided by its area.

The piston B, it will be observed, is exposed to gas pressure below and spring pressure above. If the former is the greater the piston is lifted by the pulsating pressure in the engine cylinder, but when the spring pressure is increased gradually, the pulsation is just stopped, and the two pressures (i.e. the compression and

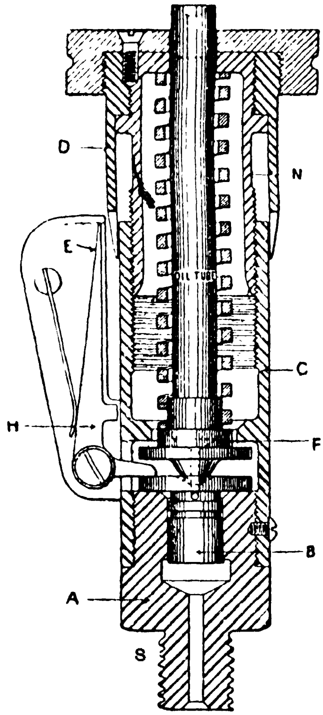


FIG. 176.—The Okill pressure indicator.

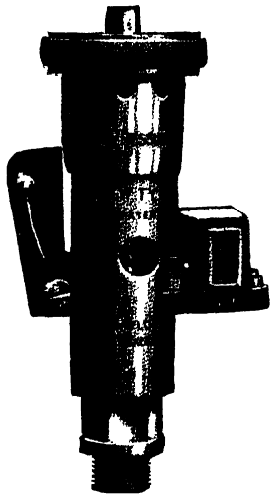


FIG. 177.

spring pressures) become equalized. The corresponding value of the pressure is then read off on the " counter " indicator.

In order to magnify this pulsation, and to serve as an indicator of pressure-balance conditions, a cranked elbow lever E is provided. The upper end moves over a fixed scale. The milled screw is adjusted until the movement of the piston B, as indicated by the lever E, just ceases.

For measuring compression pressures, the engine must be motored around, if a single-cylinder type, but for multi-cylinder engines, the sparking plug of the cylinder under test is short-circuited.

For explosion pressures, measurements are made whilst the engine is running in the ordinary way.

Research Type Indicator.—In the case of the Okill research type pressure indicator a number of useful refinements have been introduced. These include an electrical method of indicating the correct pressure value, a water-cooled barrel, and forced lubrication to the piston.

Fig. 178 shows a sectional view of this instrument. The method of using it is as follows : connect the indicator through a cock screwed into the cylinder. Open the cock and adjust the compression of the piston-loading spring until the electric lamp, connected across contacts on the vibrating tell-tale pointer and its bracket, lights continuously. Now release the pressure until the lamp just flickers. The adjustment of the knurled head which controls the pressure of the spring on the piston should be such that the slightest rotation in opposite directions causes the lamp to flicker or to burn continuously. The correct pressure can then be read off.

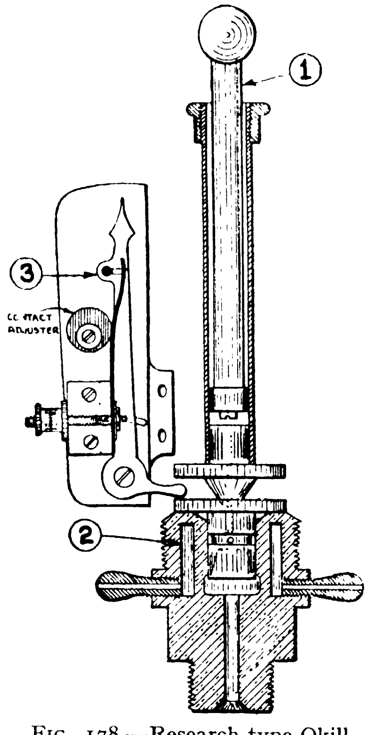


Fig. 178.—Research type Okill indicator.

The barrel of the cylinder is water-jacketed, an efficient circulation being obtained from a one-half gallon water-tank placed at a height of 3 to 4 feet above the indicator ; two rubber tubes connect the indicator to the tank.

The detachable plunger of the indicator forces lubricating oil from the hollow piston-rod into a groove cut in the indicator piston, the indicator cock being *closed* when the oil plunger is being used. The incorporation of the water-cooling and oil-feed to the piston enable this pattern of pressure indicator to be used over relatively long periods.

Diesel Engine Pressure Indicator.—Another model of the Okill pressure indicator, of the geared-nut type, is now made for indicating pressures up to 4000 lb. sq. in. This instrument is particularly suitable for the measurement of fuel injection delivery pressures and the pressure in the pipe line when the fuel valve is open and closed in the case of Diesel engines.

In addition, there is a high-pressure indicator valve for Diesel engines.

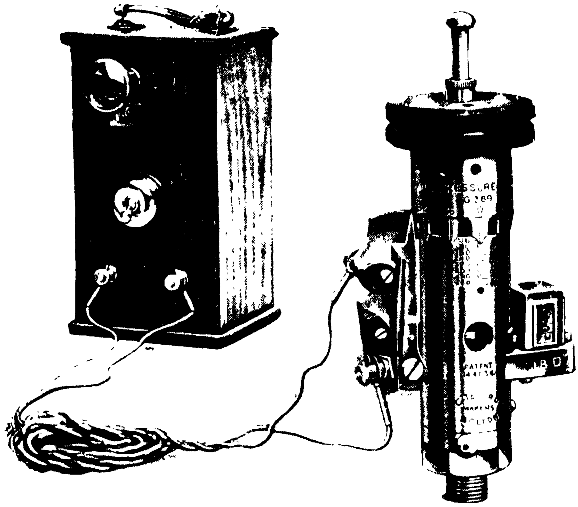


Fig. 179. Research type Okill pressure indicator.

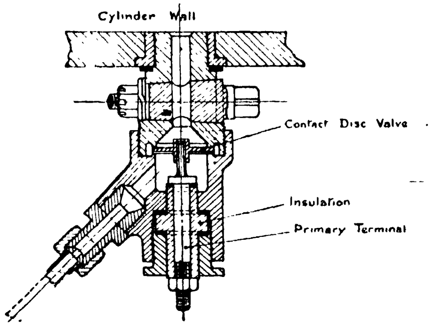
[*To face page* 230.

The (Dobbie-McInnes) "Farnboro" Indicator.—Although designed and intended, primarily, for cyclical-pressure and valve displacement variations, this indicator is most convenient for measuring compression and explosion pressures, a special gauge device being provided for this purpose.

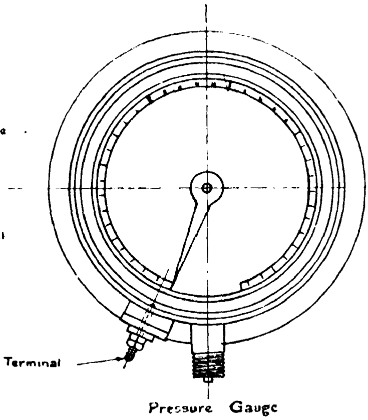
It is proposed, therefore, to describe the general principle of the indicator here, from the point of view of single-pressure measurements, and the complete cyclical-pressure instrument later.

The principle employed consists in balancing the cylinder pressure on a small disc type valve, placed very near to the combustion chamber, by means of air pressure on the opposite face of the disc, and of indicating the balancing-pressure value electrically.

Referring to Fig. 180 A, which illustrates a section through the balance valve unit, A is the contact disc valve (of stainless steel) which is exposed on its upper side to cylinder pressure, and on its



A.



B.

FIG. 180.—Balance valve unit, and gauge.

lower side to pressure from a compressed air supply. The valve is free to move up or down by a small amount, namely, about 0·01 inch, so that it can float between the two faces shown. It is guided by a spindle which is insulated, electrically, from the rest of the body. The spindle guiding the disc is connected to the primary circuit of a high-tension coil, and the two faces between which it floats form the "earth" of this circuit. The disc, therefore, acts as a contact breaker for this electrical circuit. If, therefore, the air pressure is measured when the disc is balanced, as indicated by the occurrence of sparks in the secondary circuit of the high-tension coil previously referred to, we have a means of measurement of the cylinder pressure.

A convenient form of maximum pressure gauge which has been designed for use with this balance valve unit is that illustrated in Fig. 180 B.

It consists of a Bourdon type pressure gauge, modified in such a manner that its pointer forms the earth of the high-tension circuit, its extremity being separated by a short-air-gap from a brass bevelled concentric annular ring which is insulated from the remainder of the gauge and is connected to the high-tension terminal of the induction coil. This air-gap acts as a spark-gap when the primary circuit at the balance disc valve is broken, and, therefore, when a "balance" between the cylinder pressure on the one side of the disc and the air pressure on the other side is obtained, a shower of sparks occurs between the pointer B, and the annular metal ring, across the air-gap. If the gauge is connected to the air-supply pipe (to the balance valve) the reading of the pointer B when sparking just begins to cease with slightly increasing air pressure will give the maximum cylinder pressure existing during the test.

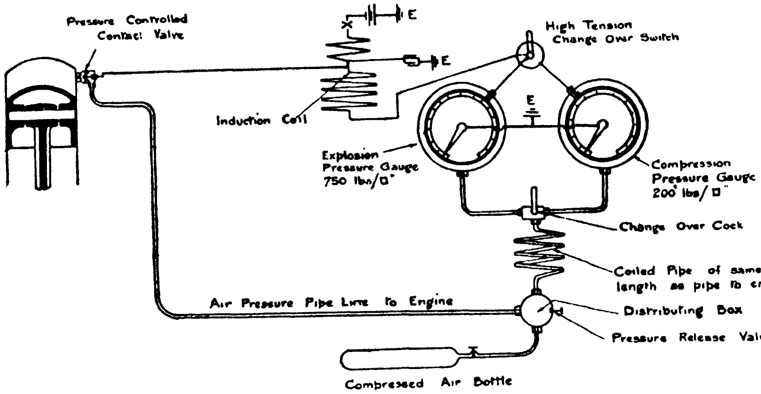


FIG. 181.—General arrangement of maximum pressure apparatus.

If the cylinder is not firing it will give the compression pressure; if firing, the explosion pressure.

Fig. 181 illustrates the general arrangement of the apparatus, and shows also an additional pressure gauge and a change-over cock. The 200 lb. (maximum) gauge is for giving a good compression pressure scale, and the 750 lb. (maximum) gauge, for explosion pressures; one gauge is, therefore, used for each purpose.

It is necessary to insert a length of air-supply tubing equal to that of the pressure pipe-line to the engine, between the distributing box shown and the pressure gauges. This form of single-pressure indicator has been very successfully employed in connection with the measurement of detonation and pre-ignition pressures upon aircraft engines. Table XIV illustrates some typical results thus obtained.

When it is desired to measure the maximum pressures of a number of cylinders the gauge high-tension terminal is connected to that of a step-up induction coil, which is placed in circuit with

TABLE XIV

Results of Tests upon Aircraft Engines

Engine	Compression Ratio	R.P.M.	Magneto Timing	Explosion Pressure (Gauge)		Compression Pressure (Gauge)
				Average	Maximum	
Dragonfly	4·4	1650	30° E.	330	—	86
Siddeley Lynx	5·0	1700	--	390	—	112
Benz	5·0	1400	15° E.	375	—	110
,,	5·0	1400	28° E.	460	—	110
Falcon	5·15	2200	28° E.	—	—	—
Liberty	5·3	1650	27° E.	500	760	120
Maybach	5·78	1400	33·5° E.	435	630	137
,,	5·78	1600	33·5° E.	400	600	135
,,	5·78	1800	33·5° E.	385	620	132·5
,,	5·78	2000	33·5° E.	350-430	630	130
Benz	6·0	1700	25° E.	510	800	145
Monocylinder, Alcohol injection	10·37	1650	16° E.	580-840* / 380-420†	950	245

* Refers to maximum power conditions.
† Refers to weak mixture.

a 6-volt accumulator and a multi-way switch. From the switch leads are taken to disc valve units screwed into the engine cylinders. Compressed air is led through a distribution box to the disc valve

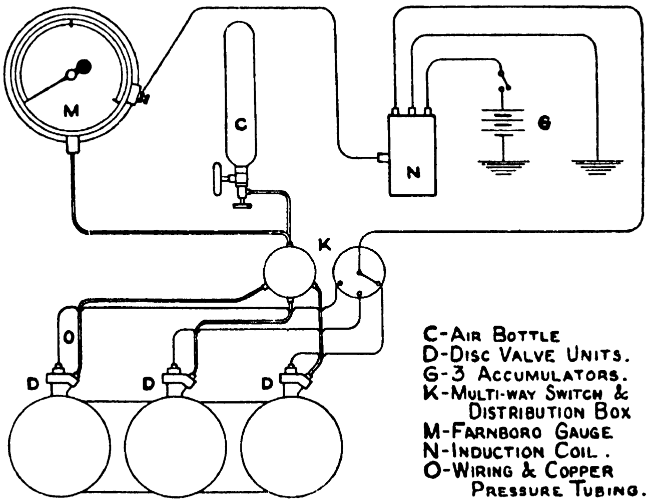


FIG. 182.—Showing the "Farnboro" pressure-gauge connections.

units, and also through tubing of the same length to a $\frac{3}{8}$-inch gas connection on the gauge.

Fig. 182 shows, diagrammatically, the pressure-gauge connections.

The Maximeter Maximum Pressure Device.—The maximeter device developed by Tchang-de-Lou and J. R. Retel of France uses the same principle as the "Farnboro" maximum pressure gauge, but arranges for the cylinder "balanced disc" to *break* an electric circuit having a lamp in series, when the gas pressure exceeds the applied air pressure. Thus, as long as the cylinder pressure is below that of the air pressure the lamp remains alight, but it goes out when the cylinder pressure exceeds the air pressure on the upper face of the balanced disc, and thus indicates when the maximum cylinder pressure is attained. When this occurs

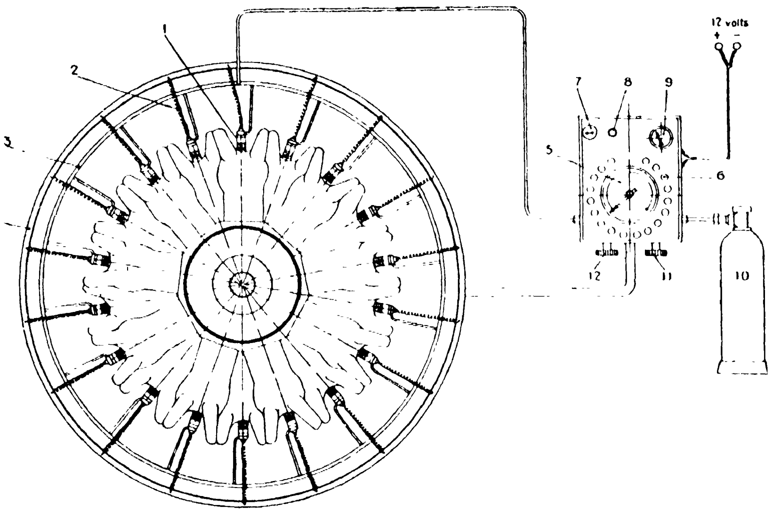


FIG. 183.—Maximeter installation on radial aircraft engines.

Key.—1, maximeter; 2, electric wires; 3, gas manifold; 4, wiring harness; 5, pressure gauge; 6, indicator lamps; 7, electrical interruptor; 8, signal light; 9, high-range pressure gauge; 10, gas bottle; 11 and 12, pressure-adjusting finger wheels.

the reading of the pressure gauge in the compressed air circuit, to the pressure unit, is read off.

Fig. 183 illustrates the application of this device to each of the cylinders of a radial aircraft engine. Each maximeter has its own indicating lamp, and all the lamps are arranged around the face of a pressure gauge so that the maximum pressures in the various cylinders can readily be checked.

Disc Valve Indicator.—An interesting form of maximum pressure indicator devised by the N.A.C.A.[1] is shown in Fig. 184. The principle is the same as that of the "Farnboro" indicator, except that no electrical apparatus is used.

The balanced and trapped pressures are recorded with a disc

[1] See footnote, p. 227.

type indicator, an auxiliary air pressure (obtained from the air bottle shown) being used to ensure pressure in excess of the cylinder pressure to be measured.

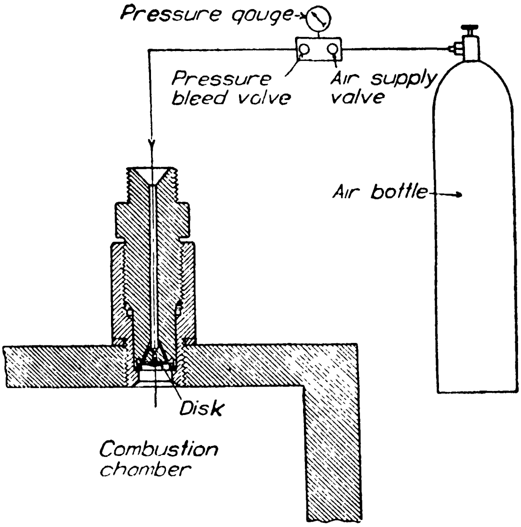


FIG. 184.—The N.A.C.A. disc valve pressure indicator.

The air pressure is admitted to the outer side of the disc and so regulated that the maximum cylinder pressures are balanced as indicated by a pressure-gauge needle. The gauge needle will fluctuate when the external pressure is less than the maximum cylinder pressure, for there will be a pressure wave produced in the line when the disc is just lifted off its seat. With the trapped pressure method the gas in the engine cylinder is allowed to lift the disc, and some of the gas that passes through the seat is trapped above the disc when it reseats. There is thus a pressure built up in the external line in communication with the pressure gauge, which is lower than but indicates the maximum cylinder pressure.

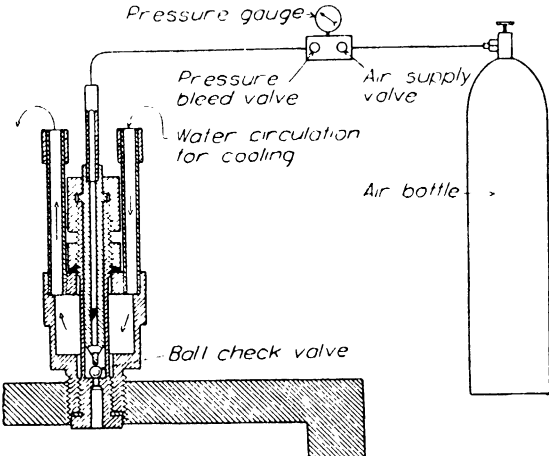


FIG. 185.—Ball-check valve-type pressure indicator.

This type of indicator has given consistent readings when used for recording by both the balanced and trapped pressure methods. In this case the disc valve has a large area, for its mass, exposed to the cylinder pressures, and has a seat width of less than 0·005 inch.

Electrical types of maximum pressure and detonation indicators,

using telephones, have also been designed, but these have not hitherto proved satisfactory. With these devices the attainment of high cylinder pressures or detonation is arranged to just cause a series of clicks in the telephones. This method requires very careful training to separate the detonation from other engine noises.

A carbon-pile indicator has been tested for maximum cylinder measurements. In this instance the diaphragm in the cylinder head communicated with the carbon disc pile by means of a light rod; the carbon pile was connected in series with a battery and headphones. The method was not very satisfactory, however, as the headphone sounds were masked by other engine noises.

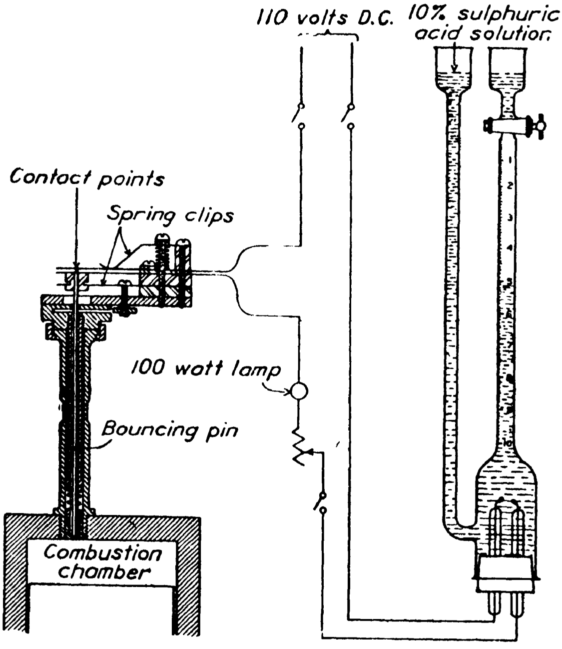


FIG. 186.—The bouncing-pin detonation indicator.

Another method, based upon the principle of the balanced pressure one, is illustrated in Fig. 185. It employs a ball-check valve between the combustion chamber and a pressure pipe communicating with an auxiliary air-pressure reservoir. Although this form of pressure indicator gives consistent readings by the balanced and also the trapped pressure methods there is a small error due to the seat width giving a different area above and below.

The Bouncing-pin Detonation Meter.—This indicator, as mentioned earlier in this book, was designed in order to show when the detonation point is reached under conditions of variable compression ratio and combustion characteristics. It has been used

in connection with fuel detonation experiments. The principle of this indicator is shown in Fig. 186.

The piston element in the combustion chamber head has a light metal pin resting on it. Any appreciable vertical movement of this rod, such as would be caused by detonation, causes it to close an electrical circuit, by forcing the two contact points together. When the circuit is thus closed, the electric current flowing is arranged to electrolyse an acidulated water solution, the volume of the gases generated being proportional to the length of time the contact points are closed, i.e. to the period of detonation.

In this case it is assumed that the amount of throw is proportional to the intensity of the detonation pressure.

Tests made by the N.A.C.A. showed that the bouncing-pin detonator did not give consistent results over a range of variable compression ratios, although at low compression ratios it gave comparable results. It was possible, at engine speeds of 1500 r.p.m., to make the pin bounce and record by compression alone when the compression ratio was raised above 7·3.

Explosion Recorders.—The ordinary cyclical-pressure type indicator gives information concerning the pressure, for one or a limited number only, of such cycles, and does not give a continuous record of the number of such cycles occurring during a test. This number can, of course, be assumed from the revolution indicator and time readings, but it presupposes no misfiring or similar irregularity to occur.

Although occasional misfiring will not affect such results as power measurements, it is often convenient, more particularly in the case of slower running engines, such as Diesels and other crude-oil types, to have a record of the total number of explosions, and of the maximum pressures developed at each explosion. In the case of engines governed on the " hit-and-miss " principle, it is necessary to have a record of the *idle* as well as the *power* strokes.

Pressure Sampling Valves.—A method which is sometimes adopted for obtaining the pressure value in the cylinder at any given point of the piston's stroke, consists in connecting the combustion chamber with a small indicator cylinder and piston through the intermediary of a rotating valve running at engine speed and provided with a small slit. The engine-cylinder pressure is then communicated to the indicator cylinder, once in every cycle, and at the same point in the cycle. This idea is elaborated in the Gale and Juhasz indicators, means being provided for varying the relative position of the slit, or its point of communication with the engine cylinder, so as to give a complete cycle of pressures which is recorded on an ordinary pencil type of indicator drum.

The inlet pressure, in the case of an automobile engine, has been measured, by a somewhat similar device to the one described,

by the arrangement [1] illustrated in Fig. 187. In this case the piston uncovered a small hole drilled through the cylinder wall, and closed it a few degrees after the closing of the inlet valve. In

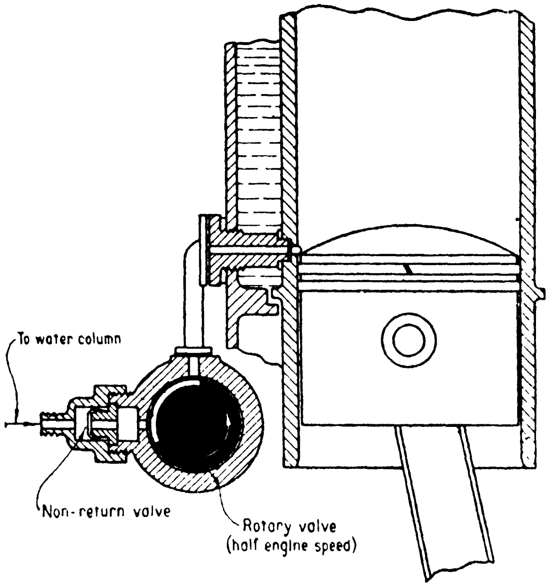


FIG. 187.—Pressure sampling valve.

the case of a poppet valve engine this hole is also uncovered during the exhaust stroke, and, since the arrangement for measuring these low inlet pressures might be deranged by the exhaust pressures, the rotary type of valve shown, driven at one-half engine speed, was

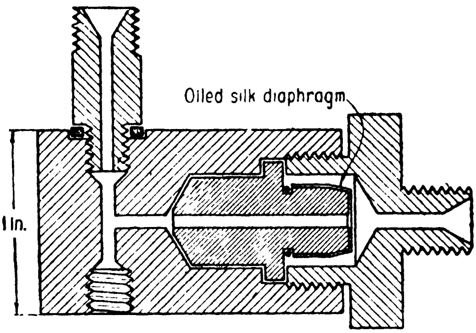


FIG. 188.—Non-return valve used with pressure sampling valve.

employed; communication during the exhaust stroke was thus interrupted. The rotary valve communicated with a delicate non-return valve of the type [1] shown in Fig. 188, in which an oiled-

[1] *Vide* "The Charging of Two-Stroke Engines," W. Morgan, *Proc. Inst. Autom. Engrs.*, vol. 17, 1922.

silk diaphragm was used to prevent any flow of the gases, and to communicate the pressure to a water manometer. With a sleeve valve engine the rotary valve is unnecessary, as the holes in the sleeve and cylinder wall can be arranged to act in a similar manner.

Sampling Valve Used with Low Speed Indicator.—The pressures obtained from the sampling valve can, as previously stated, be used in conjunction with a low speed steam-engine type of indicator for obtaining average pressure-volume diagrams for a number of consecutive cycles.

A pressure-sampling indicator of this type, devised by F. L. Prescott of the Wright Field, Dayton, Ohio, U.S.A.,[1] employs the sampling valve shown in Fig. 189. It consists of a poppet valve,

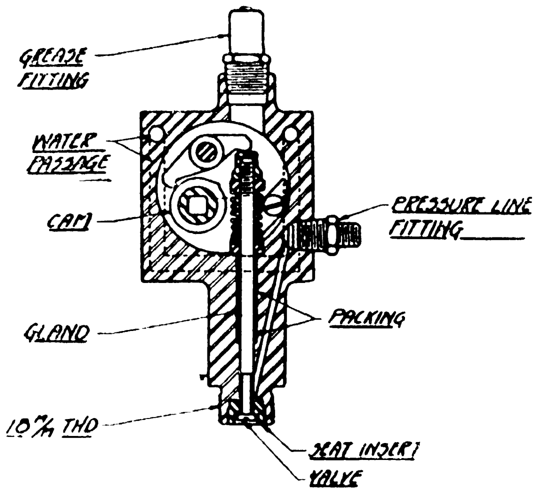


FIG. 189.—The Prescott pressure-sampling valve.

spring, tappet nuts, rocker and cam. The cavity in which the cam and rocker operate is charged with cup grease.

The valve proper is made of steel and has a lift of 0·007 inch. Above the head the stem is reduced in diameter, so that a degree of pressure balance is secured. The clearance so formed also acts as the port from which gas pressure from the engine cylinder is transmitted to a pressure-line fitting in the side of the body.

The stem of the valve is sealed in its guide by packing to which oil is supplied under pressure. A line from the engine lubricating system can be connected to a fitting for this purpose at the cover plate which covers the assembly. This sampling valve is screwed directly into the cylinder by means of an 18 mm. thread. The sampling valve can be water-cooled if desired.

[1] *Autom. Industries*, March 25, 1933.

The sampling valve is driven through a light shaft and universal joints by the phase gear, which is also shown in Fig. 190. This unit serves to open the sampling valve for an instant, at progressively later points in the engine cycle. It comprises two spur gears—a pinion driven from the engine, usually through chain and sprockets, and an internal gear which has its bearings in an eccentric carrier. The carrier is rotated by a hand-wheel and worm, to vary the angular relation of the gear to the pinion through 360°, thus changing the phase of the sampling valve through a whole engine cycle.

On the outer end of the eccentric carrier is a cord drum, which is calibrated in degrees to serve also as an index to phase change. The cord from the steam-engine indicator is attached to the cord drum of the phase gear, so that the former acts as recorder of pressure

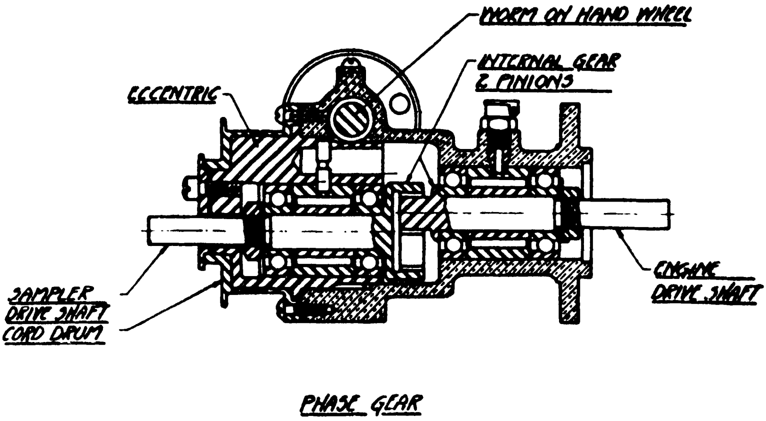


FIG. 190.—The Prescott phase alteration gear.

changes sampled at the engine cylinder and transmitted to it through a copper tube of $\frac{1}{16}$-in. inside diameter.

Cyclical Pressure Indicators.—The purpose of these indicators is to give a pressure variation diagram over the whole or part of a working cycle; in some instances several consecutive cycle diagrams may be recorded.

For the purpose of mean indicated pressure and indicated horsepower measurements it is necessary to obtain pressure-volume diagrams, i.e. diagrams having pressures as ordinates on a piston-stroke base. For observation purposes these diagrams are useful for noting the changes that occur as a result of varying the engine speed, mixture strength, throttle opening, ignition advance, etc.

For visual observations or records of rapid pressure changes, namely, during the actual combustion period, pressure diagrams on a crank-angle (or time base) are much better than those on a piston-

stroke base, although the displaced or "out-of-phase" pressure diagrams in the latter case are very useful. In all cases, indicators consist of two principal units, namely, (1) a pressure element and (2) a displacement unit; the latter gives the crank-angle or piston-stroke base for the pressures recorded by (1).

Types of High Speed Indicators.—It is not possible to devote space to a review of the development of the high speed internal combustion indicator into its present forms, but to those who are interested the footnote references [1] may prove useful. It is only possible here to describe some of the more successful modern indicators. Indicators of the present time may be divided broadly into four main groups or classes, as follows:—

1. *Mechanical Types*, employing pencils or styluses. These also utilize the sliding piston and straight line mechanisms.

They represent a development from the slower speed explosion

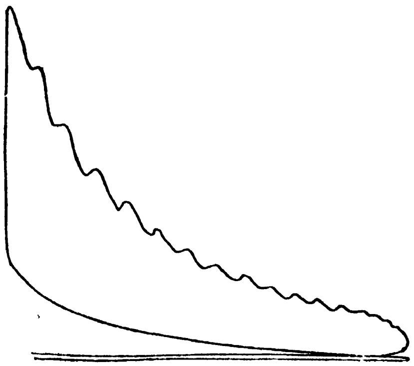


FIG. 191.

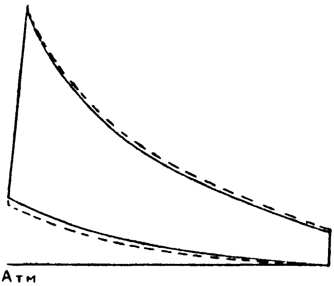


FIG. 192.

engine type indicators, in which the disadvantages, such as inertia, pencil friction, flexure, and spring vibration, have been overcome more or less satisfactorily.

The ordinary mechanical indicator is seldom if ever used for high speed engines, being suitable only for speeds well below 1000 r.p.m.

The inertia effect due to the weight of the piston is such as to produce erroneous indicator diagrams and the combined effect of inertia and frequency of vibration of the control spring is to give rise to synchronous vibrations of the pencil. A typical result is a wavy expansion line, as shown in Fig. 191.

Friction of the pencil on the paper record also gives an incorrect diagram as the pencil tends to lag below the compression line and above the expansion line as indicated in Fig. 192. At higher speeds

[1] "Indicators" (a Symposium of Papers on Indicators), L. Pendred, *Proc. Inst. Mech. Engrs.*, January, 1923.
"The Optical Indicator," W. Morgan and A. A. Rubbra, *Proc. I.A.E.*, vol. 21.
"Some Applications of the Indicator Diagram," R. W. J. Pryer, *ibid.*

friction of the pencil often causes flexure of the pencil arm which introduces yet another source of error.

Modern mechanical type indicators include the Micro indicators of Dr. Mader, and Collins, in which the reduction in dimensions of the moving parts is such that microscopic diagrams, free from most of the usual errors, are obtained. These are magnified, or viewed through a special measuring microscope.

The Dobbie-McInnes Super Diesel explosion engine type indicator gives a diagram of only 2 by 1 inch. Another example is the small "Crosby" type indicator supplied by The Lunken Indicator Company, which is stated to be suitable for speeds up to 1000 r.p.m., has a recording drum of $\frac{3}{4}$-inch diameter, and gives a diagram whose largest dimensions are $1\frac{1}{2}$ by $\frac{15}{16}$ inch.

2. *Optical Indicators.*—The general principle adopted in these is to replace the piston and pencil mechanisms by an optical mirror or mirrors, which are given only a rocking motion, instead of one of translation. In this manner the inertia forces are reduced to exceedingly small dimensions. In some cases one mirror only is employed, and is rocked about two axes by the pressure and piston phase components respectively. Examples of this type are the Schultze, Hospitalier Charpentier, and Hopkinson optical indicators.

Another set employs one mirror for the pressure motion and a separate one for the piston position motion, a beam of light being reflected from the source, off the former (usually) to the latter, and thence on to a ground-glass screen, or photographic plate. Examples of this type include the original Watson, and subsequent Watson-Dalby, Burstall, and Midgeley indicators.

Apart from the mirror arrangements, optical indicators also differ in the pressure communication means. Some indicators, such as the Watson, Watson-Dalby, Schultze, and Hospitalier Charpentier types, employ circular sheet-metal diaphragms clamped around their edges, and allow these to act both as the pressure element and the spring control; others employ special types of pistons for the pressure element.

Diaphragms.—If flat diaphragms are used, the pressure- (or load) deflection curve of calibration is not a straight line one, whereas, if a corrugated section disc be employed, as in the Watson-Dalby type, the scale is practically a linear one. Diaphragms are usually made of spring steel, such as silicon or chrome-vanadium steel, or of stainless steel. Owing to the presence upon the one side of the hot, moist gases, the corrosion effect on the steel diaphragms is a serious item. Professor Watson overcame this difficulty by gilding the discs; the use of chromium-plated steel or of stainless steel is also a satisfactory solution.

One drawback in the use of diaphragms is the dependence of the pressure scale upon the temperature of the diaphragm, the

stiffness decreasing with temperature increase. Since the diaphragms are generally calibrated in the "cold" condition, a possible error arises in practice. It is not a difficult matter, however, to devise means for calibrating the discs at temperatures similar to their working ones. In the Watson indicator the diaphragm was held in a hollow chamber, through which a continuous supply of water passed. The clamping of the diaphragm is also an important item. Unless this is firmly held serious errors in the pressure—deflection scale—will occur, as shown in Fig. 193 by the dotted lines.

The hysteresis of the diaphragm is also a further source of error as, owing to the rapidity and the high frequency of the loading, the metal probably never accurately follows the true static loading curves. One effect of this is to cause a "lag" when the pressure is coming off, during the expansion stroke; the expansion curve is then higher than it should be.

A better arrangement, it would seem, would be to employ a very thin diaphragm to act as a gas-tight pressure element with a separate external spring of the cantilever or beam type. In the Smith indicator the pressure was received on a light copper disc, another spring steel type of disc being used as the spring control.

The Hopkinson and Burstall indicators, special pistons with external springs of the cantilever type, are employed.

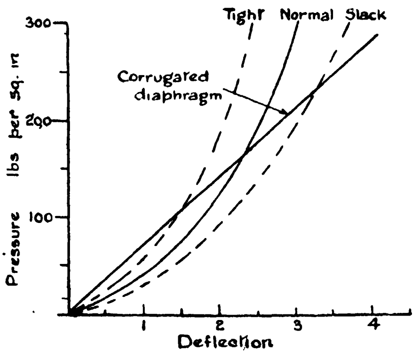


FIG. 193.

Optical Indicators in General.—It has been stated that the optical type of indicator possesses the marked advantage of being able to reduce inertia forces to a negligible amount; further, it eliminates pencil friction. The optical indicator is usually a rather delicate piece of apparatus, bulky in form, and requiring a skilled operator. It is not yet sufficiently portable for ordinary commercial use, nor would it generally stand up to the rough usage in the test-house or workshop for long periods. On the other hand a skilled operator having a knowledge of the physical principles involved should be able to obtain accurate indicator diagrams of relative large size and at speeds up to nearly 2000 r.p.m. which are generally better than those given on the curved base screen of the present type of cathode ray type indicator. Further, the diagrams can be observed or photographed with equal facility.

In this connection the writer has observed and photographed a considerable number of optical indicator records of remarkable accuracy and detail. Very small changes in the adjustments of an

engine are at once revealed on the ground-glass screen by alterations in the diagrams which ordinary indicators mask.

The use of light-tight indicator boxes, and of photographic films or plates is a drawback for commercial routine tests; the fact that each diagram represents not the average, but one, only, of the very large number of cycles is also regarded as a disadvantage. By watching the image on the ground-glass screen, however, it can usually be observed when conditions are constant, and the diagrams taken at such times, and at regular intervals.

For laboratory, technical instruction, and engine research purposes, the optical indicator of to-day is probably the best of any available. The fact that the diagram is always visible, and that changes in the throttle carburettor, ignition, and similar items can be followed on the ground-glass screen makes this indicator particularly useful for instruction purposes, and also for original investigations.

3. *Miscellaneous Types.*—In the endeavour to avoid the disadvantages of the two general types already mentioned, indicators have been evolved which depend upon quite different principles.

The most notable and successful of these types is the (Dobbie-McInnes) "Farnboro" indicator, the principle of which, as we have already shown, depends upon balancing the cylinder pressure, which acts on the one side of a disc, by means of air pressure on the other side of the disc, the moment of balance being indicated by electrical means. The air pressure value at the balancing points is recorded on a rotating drum at the correct piston or crank phase.

The "balanced" pattern is not the only form possible, and now that very sensitive galvanometers are available, it is possible that the purely electric indicator will come into the foreground.

Professor Trowbridge [1] has devised an indicator in which a very small diaphragm is employed. A very small coil of fine wire is attached to it, and into this coil one pole of an electromagnet projects. When the coil is moved it generates an electric current, the E.M.F. of which is proportional to the *velocity* with which the disc moves.

The current is taken to an Einthoven type galvanometer, the mirror of which reflects a beam of light on to a photographic film. By means of a commutator on the crankshaft a record can be taken at any part of the stroke. The pressure at the point selected is taken as being proportional to the velocity of the disc, to which, as we have stated, the current generated is proportional. The advantage of this method is that the pressure unit is very compact and can be separated to any extent from the indicating unit. The former can be screwed directly into the cylinder head.

[1] Of Princetown, America.

This instrument gives a series of pressure plottings upon a "crank-angle" or "time" base.

Cathode Ray Indicators.—The principal objection to most optical indicators when used at high speeds is that of inaccuracies in the diagrams caused by the inertia of the moving mirror or mirrors. This effect and in some instances inertia of the pressure diaphragm has limited the use of optical indicators to speeds below about 2000 r.p.m. With the increase in engine speeds up to 2 or 3 times this value the necessity arose for an improved design of indicator having no appreciable inertia. This requirement is now fulfilled in most of the cathode ray oscillograph type indicators, which utilize the cylinder pressure variations to give corresponding voltage or potential difference changes; the latter are then amplified and applied to a pair of control plates in a cathode ray tube of a similar type to that employed for television purposes. The stroke or crank-angle base for the cathode ray spot on its screen is obtained by suitable electrical means, as explained in Chapter 8.

Present experience with this type of indicator shows that in the hands of a skilled operator, namely, one conversant with the electrical principles and adjustments, it gives reliable results. It also has a much wider range of application to internal combustion engine problems than the optical type.

Mechanical Indicators.—(*a*) *Micro Types.*—The principle of this type of indicator consists in reducing natural vibration period of the inertia forces to small dimensions by diminishing the size of the instrument, and by employing a stiff helical cantilever, or beam type of spring. The period of vibration is reduced to something less than one-third of the ordinary value in this manner. The effect of having such a stiff spring is to reduce the pressure scale considerably, so that the vertical, or pressure ordinates are small; since, also, the movements corresponding to the piston displacements are kept small, the resulting diagram is very small. In the Mader type, the diagrams taken are included in a rectangle of sides about $\frac{1}{10}$ inch. In the case of the Collins indicator the diagrams are approximately $\frac{1}{8}$ by $\frac{1}{10}$ inch. Magnification is, therefore, necessary for examination and measurement purposes.

The principal disadvantage of the micro-indicator is that it is not possible to obtain visual observations of the pressure changes during actual tests. Moreover, the relative thickness of the pointer lines in relation to the area of the small diagram traced out is such as to reduce the accuracy of any pressure measurements made from the diagrams.

Pointer friction is another drawback with this type of indicator.

(*b*) *The Cambridge Micro-indicator.*[1]—This instrument is illustrated in Fig. 194, and diagrammatically in Fig. 195. It employs

[1] Manufactured by the Cambridge Instrument Co., Ltd., Cambridge (Collins Patent).

a notched circular celluloid disc, upon which the diagrams are finely indented or marked, by means of a suitable pointer. Ten records can be obtained upon the one disc, and means are provided whereby an automatic switch controls the recording mechanism so that it is disconnected from the record after one complete cycle; this avoids overlapping of successive diagrams.

Referring to Fig. 195, there is a removable cylinder liner W of stainless steel, within which the piston V works. This liner may be replaced by others of smaller internal area, so that a range of higher pressures is possible. The piston-rod O, which is formed solid with

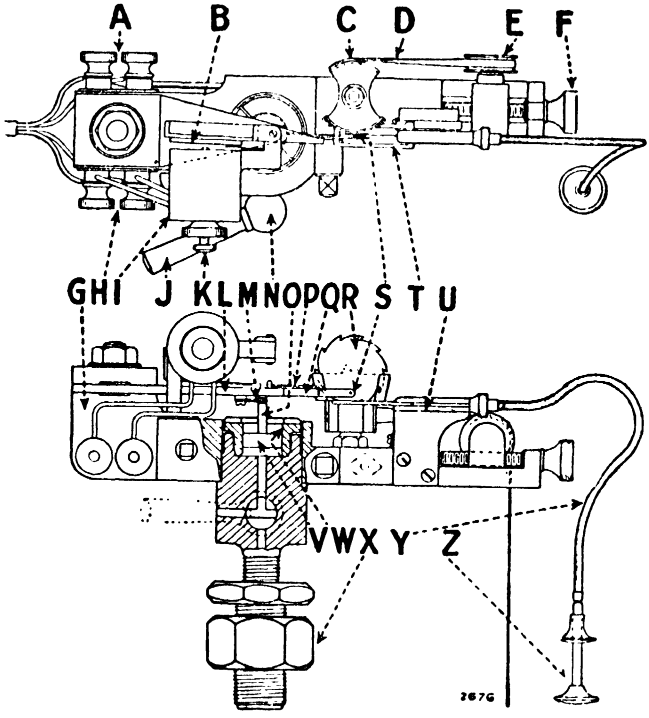


FIG. 195.

the piston, is kept in firm contact with the pressure spring L, by means of the subsidiary spring M, the former being bolted firmly to the rigid frame G. Attached to the pressure spring is a light arm Q, carrying a recording stylus S. This stylus produces records near the edge of the celluloid disc R, which is carried on, and bent to conform with the cylindrical surface of the vertical drum C, to which it is secured by means of clips and a central pin. The drum C is given a reciprocating motion about its axis from the light steel tape D, which, after passing over the adjustable guide pulley E, is attached to a crank or other suitable mechanism, synchronizing

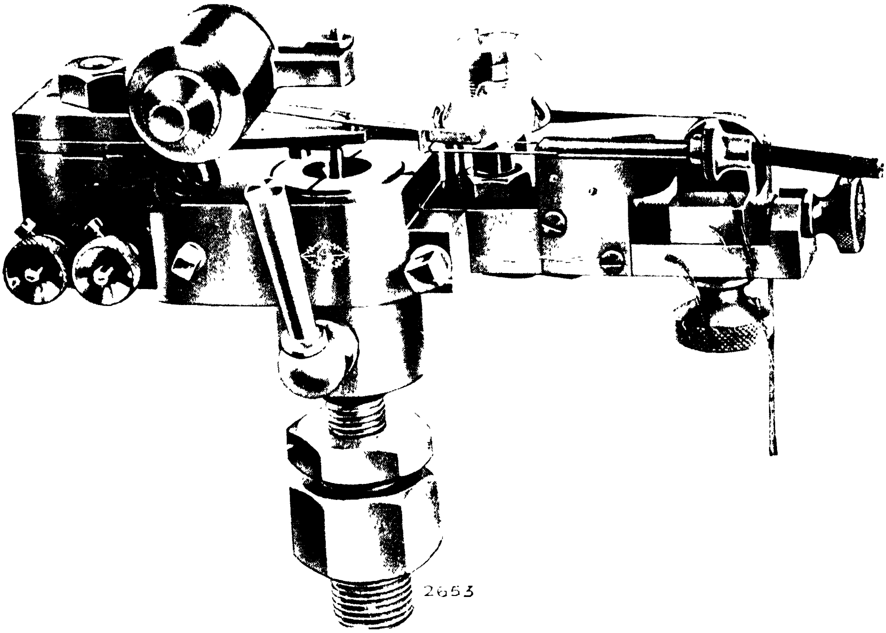


FIG. 194. The Cambridge micro indicator.

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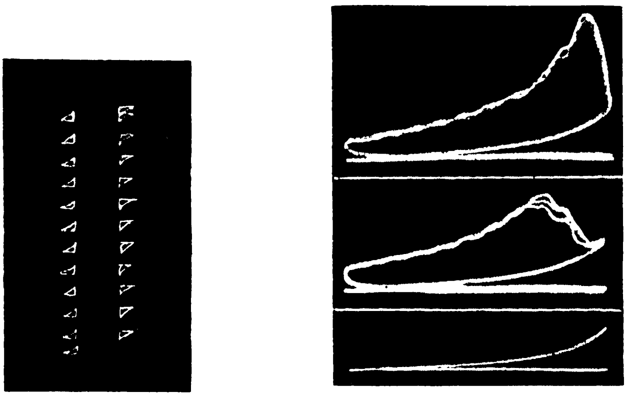

FIG. 196.—Examples of diagrams obtained. Left: actual size. Right: enlarged.

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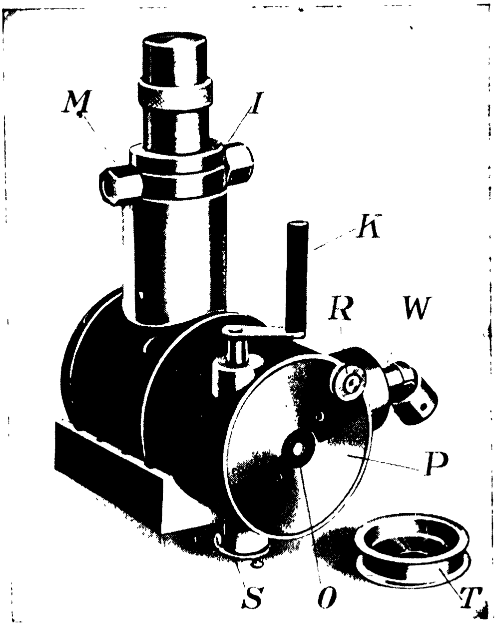

FIG. 197. The De Juhasz valve element.

[See page 247.

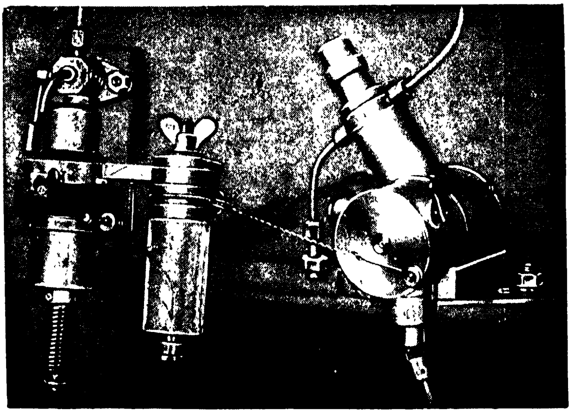

FIG. 199.—Showing the valve element, indicator drum, and driving gear on an engine under test.

[To face page 247

in movement with the engine piston and giving the periphery of the drum a travel of about 3 mm. The instrument is attached to the engine under test at any convenient point by means of the union X, carrying the tap N, which controls the communication between the engine and indicator cylinders.

To obtain a series of diagrams the celluloid disc R is rotated by means of a pawl T engaging with the notches upon its circumference, the movement of the pawl being controlled by a Bowden wire push switch Z. Normally the stylus S is not in contact with the disc R, but it may be brought into contact by the electromagnet I, which attracts one end of a light pivoted arm P, the other end of which bears against the arm Q, carrying the stylus against the face of the celluloid disc ; the pressure of the stylus is regulated by the screw K.

Reference has been made already to the automatic switch which allows only one diagram at a time to be taken. This device, which is driven at engine speed, operates through the electromagnet I, brings into action and releases the stylus from the disc R after one complete cycle.

The indicator diagrams measure about 3 mm. long by 2·5 mm. high, and consist of fine indentations made by the stylus upon the celluloid surface, being permanent in character. These diagrams are shown to actual size, and magnified in Fig. 196. The originals will bear considerable magnification. They can be examined, at once, by means of a suitable microscope, or direct enlargements from the actual diagrams can be made by photographic means, by a camera lucida, or by projection upon a screen and tracing.

The De Juhasz Indicator.—The essential feature of the De Juhasz indicator[1] is a valve element driven positively by the engine under test. This element is interposed between the engine cylinder and the cylinder of the pencil indicator unit, which latter resembles the ordinary steam or gas engine type. The valve element is connected by means of a short length of small bore tubing to the cylinder under test, so that pressure errors due to the connecting tube are minimized. The valve unit consists of two slide-valves, which reciprocate within a small cylinder, an external view of which is shown in Fig. 197. The design of the valve is shown diagrammatically in Fig. 198. Two slide-valves provided with ports reciprocate within a housing in which are similar ports, one of which is connected with the engine cylinder and the other with the indicator. In actual practice, instead of the flat slides as shown, concentric cylindrical slide-valves are employed ; these have the advantage of being easily and accurately machined. The slide-valves are driven by a crankshaft, the ports in the slides registering only once

[1] Manufactured by Lehmann & Michels, Hamburg. Supplied in this country by The Lunken Co., London.

per revolution with the ports in the housing. This is of importance, and is attained by making the angle of the two crank arms less than 180°. The main advantage of having but one communication per revolution instead of two, as would be the case had the cranks an angle of 180°, lies in the fact that the duration of communication is half as long as in the latter case. Advantage is taken of this circumstance to increase the rapidity of opening by arranging the proportions of the valve gear and the crank angles in such a manner that the ports coincide when the valves are moving at their maximum velocity. The cylinder (Fig. 197) has two ports, one of which, M, is connected to the engine cylinder, and the other, I, to the indicator unit shown on the upper right-hand side in Fig. 200. A length of metal tubing connects the valve element shown on the left-hand side with this indicator; it is stated that within certain limits the length of this tubing is not so important, and may be made appreciably longer than that of the engine cylinder connection. The slide-valves can be brought into this position by turning the hand-crank K, until the ends of both slide-valves are in line with the end of the cylinder. In all other positions the communication between the ports M and I is interrupted. The crankshaft of the slide-valves is driven positively by means of the shaft W (Fig. 200), which is engine-driven.

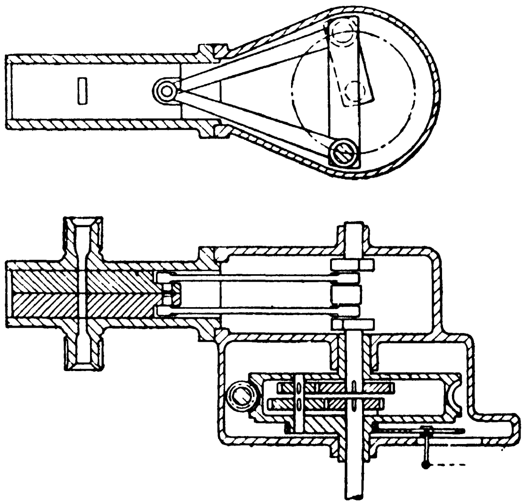

FIG. 198.—The valve element of the De Juhasz indicator.

It follows that the engine and indicator unit are only in communication in the same phase of the working cycle. In order to alter this phase, so as to obtain pressure records for other parts of the engine cycles, a subsidiary movement is imparted to the crankshaft actuating the valves. This is effected by turning the disc P; a definite rotation of this disc corresponds to a displacement of the phase by a similar amount. The phase-disc P can be actuated either automatically (through a 300 to 1 reduction gear) or by hand, using the hand-crank K.

The movement of the indicator drum is reduced from the phase-disc P. When it is desired to obtain a pressure-crank angle diagram, the indicator drum is rotated continuously by means of a suitably-

guided endless belt or cord, driven off the pulley T (Fig. 197). For the pressure-stroke diagrams the indicator drum is reciprocated about its axis by means of a cord attached to it and to the eccentrically placed stud R. For any initial adjustments the phase-disc can be rotated freely on its shaft by unslackening the nut O.

The indicator unit, which is an ordinary pencil type one, is mounted upside-down (Fig. 200). Since the speed of travel of the pencil indicator piston is only $\frac{1}{300}$ that of the engine piston speed, when the automatic phase variation gear is in use, questions of pencil and piston friction and inertia are of little importance. The indicator space above the piston is filled with oil, in order to form an air-tight seal.

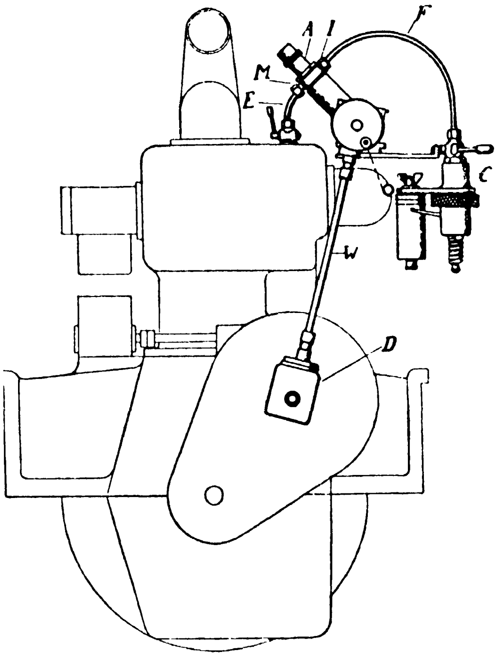

FIG. 200.—Showing arrangement of indicator elements.

The shaft W may be provided with a pair of universal joints, in order to compensate for small angular differences in the drive. The manufacturers state that the latter must not exceed 10°. It is evident, however, that any variation of angular velocity, due to non-parallelism of the initial and final driving shafts, will vary the phase incorrectly.

The indicator is provided with stiff and weak springs. Both the ordinary and the light-spring diagrams can be obtained simultaneously or consecutively.

Figs. 201 and 202 illustrate typical normal and light-spring diagrams obtained at speeds of 1000 and 3000 r.p.m. respectively.

The De Juhasz indicator has been used in the development of the Fiat supercharged racing car engine, the maximum speed at which diagrams have been taken being about 5500 r.p.m.; the maximum explosion pressures occurring during these tests were of the order 700 lb. sq. in.

Optical Indicators.—Apart from the experimental designs previously mentioned there have been several commercial types which have generally given satisfaction at low to medium engine speeds.

It is not, however, possible nor perhaps desirable to describe the various indicators, so that present considerations will be confined to an account of the original Watson indicator—which is regarded as an excellent example to illustrate the basic principles involved in optical indicators—and certain references to two or three commercial and research instruments.

It may be mentioned that several indicators of this class, including the Schultze, O.S. Manograph, Hopkinson, and Watson-Dalby were fully described in the second edition of this book.

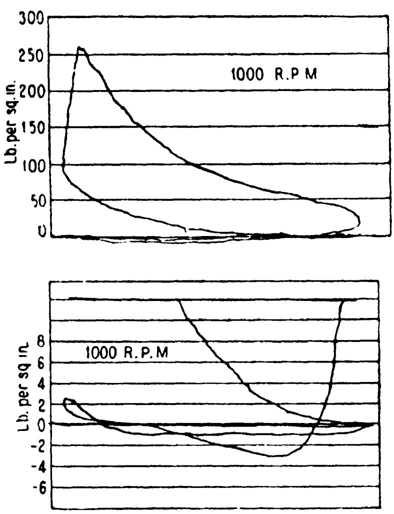

FIG. 201.—Indicator diagrams from Fiat car engine, 65 × 110 mm.

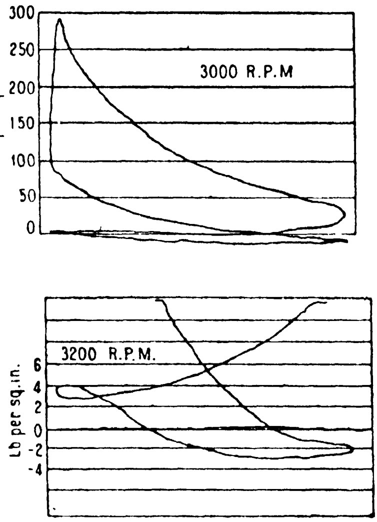

FIG. 202.—High speed indicator dia, grams from Fiat car engine- 65 × 110 mm.

FIG. 203.—Suction diagram taken from carburettor choke tube of four-stroke engine at 1200 r.p.m.

The Watson Indicator.—This optical indicator was designed by the late W. Watson, F.R.S., as a result of his experiences with the other types of optical indicator, and with the object of eliminating the defects of these types. The final instrument in the hands of physicists, was capable of yielding very satisfactory diagrams, but it can hardly be classed as a portable type of commercial instrument.

The indicator gave satisfactory results at engine speeds up to about 1800 r.p.m. above which the inertia effects of the pressure

diaphragm and moving mirror tended to introduce errors. In view of the interesting basic principles involved—and these have since been adapted in later optical indicators—a more detailed account is given of this indicator.

In the Watson indicator, a corrugated pressure diaphragm of silicon steel was used, in conjunction with a concave mirror (for the pressure motion), whilst a second and independent plane rectangular mirror was employed for the piston displacements. The diaphragm was water-jacketed to keep its temperature down to definite limits. A neat form of phase-adjustment gear was also employed, which enabled very fine adjustments to be made whilst the engine was working.

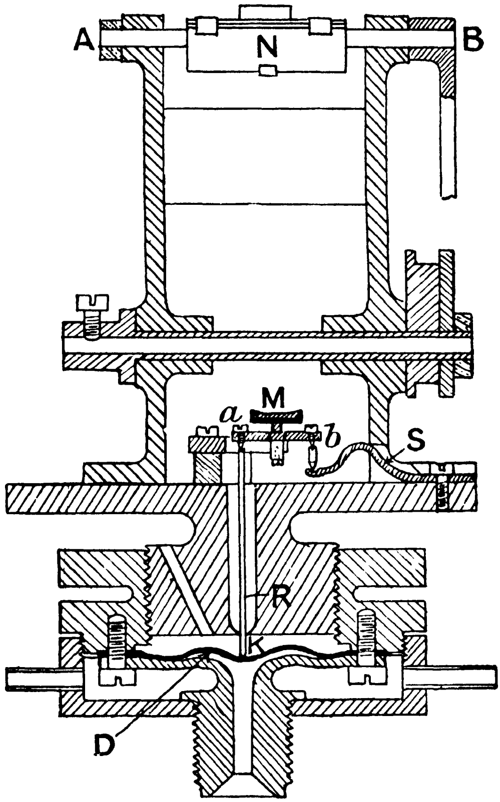

FIG. 204.—Illustrating principle of the Watson indicator.

Fig. 204, which illustrates the principle of the Watson indicator, will serve for the purposes of explanation.

The lower communicating passage to the cylinder will be observed; this leads to the lower side of a gilded spring steel disc D of corrugated section. The object of the corrugations is to enable a uniform pressure scale to be obtained.

The pressure on the diaphragm D is transmitted by means of a light, ball-ended rod R to one end of a light beam *ab*, which is pivoted at its centre, under the concave mirror M. At the end *b* is a small strut with pivoted ends, held in place by means of a light steel spring S. The mirror M is thus rocked about a central fulcrum or pivot, by the pressure motion.

A rectangular plane mirror N is provided for the piston motion, and is rocked about its axis AB by means of an eccentric and rod, the end of the latter being shown at B. Below B is shown the eccentric (in section), which is attached rigidly to the hollow shaft below and parallel with AB.

A beam of light is reflected off the concave mirror M, whence it receives a rocking motion proportional to the pressure, and thence on to the mirror N, which adds to it a motion which is at right angles to the pressure one. The beam of light with its pressure and piston displacement motions is then reflected (and, as will be shown later, focused) on to a ground-glass screen for visual observations, or for photographic records on to a photographic plate.

The optical arrangements of the indicator are shown in Fig. 206. In the original instrument an electric arc A was used as the light source, a convex lens B focusing the light on to a small hole in a diaphragm at C; the size of this hole was variable. From C the light diverged on to the pressure mirror M, whence, after reflection off the piston displacement mirror N, it was brought to a focus on

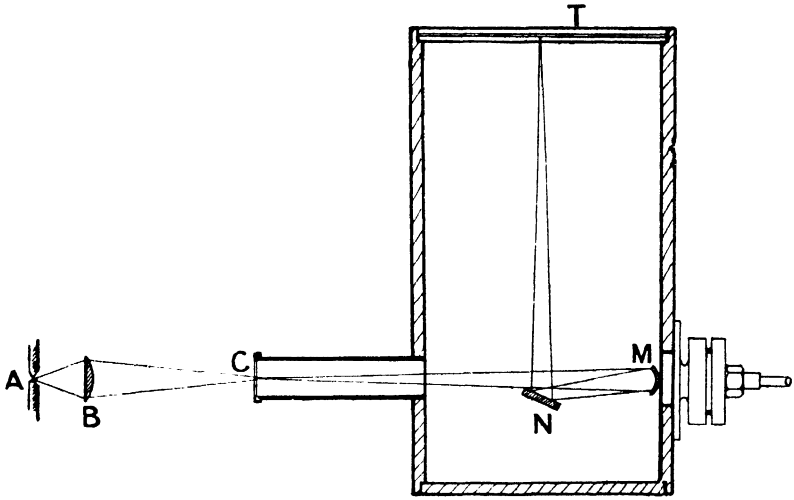

FIG. 206.—Optical arrangement of Watson indicator.

the ground-glass screen T. The points C and T were thus conjugate foci of the concave mirror N. With this arrangement, and using an arc as the source A, a very bright image was obtained at T, and the resulting indicator diagram could readily be observed in ordinary daylight on the screen.

The piston displacement phase-adjustment gear was also very neat. The chain wheel shown on the left-hand side (Fig. 207) was driven off the engine crankshaft, and drove the bevel gear E, which, in turn, drove the co-axial bevel gear F, through the intermediary of the bevel D. This latter bevel rotated about an axis, which could be moved bodily. The bearing for D was carried on the bracket P, which could be rotated through an angle of 180°, about the common axis of E and F. The bracket P could be moved either by definite angular amounts, or could be adjusted to minute amounts by means

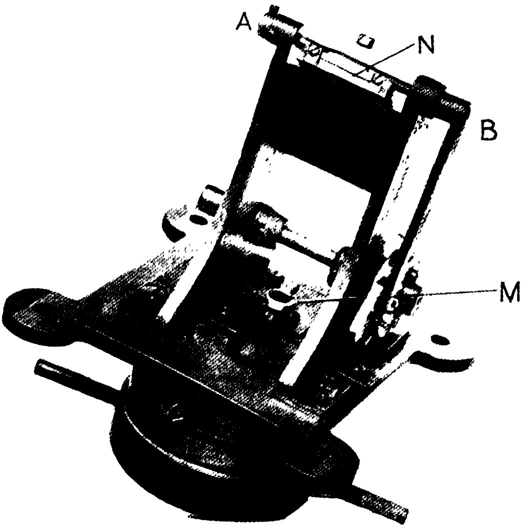

FIG. 205. External view of Watson indicator, showing concave mirror M and plane mirror N

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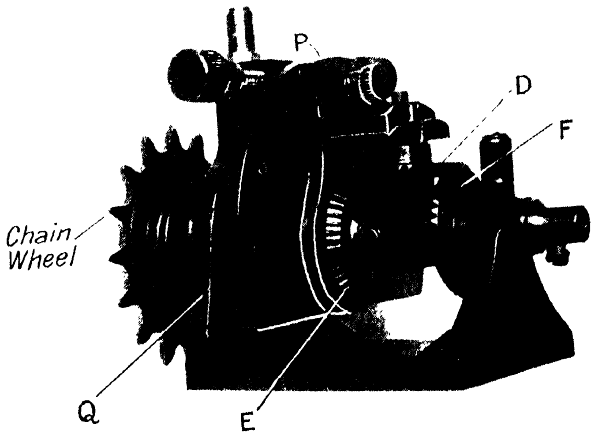


FIG. 207.—Phase gear of Watson indicator.

[See page 252.

of the tangent screw shown to the left of the letter P (Fig. 207). In this manner the position of the shaft of wheel F, which was connected to the eccentric of the indicator (for rocking the mirror N), could be displaced by any given amount relatively to the driving shaft of wheel E, i.e. the engine shaft.

This device enabled the phase of the indicator piston displacement motion to be varied relatively to that of the engine.

This adjustment is essential in all types of indicator to obtain synchronism between the two, and also for obtaining " displaced " diagrams, such as those for studying pressure rates.

The phase-gear shown also enables the indicator to be used for a number of cylinders of a multi-cylinder engine in turn. The principal advantages of this design of indicator are that (1) the piston and pressure motions are independent ; (2) the pressure scale is uniform ; (3) an exceedingly fine adjustment of phase can be obtained ; (4) the disc is water-cooled, and has a very small clearance ; and (5) the connecting tube to the tap and engine is short and free from " bore waves."

Experience with this indicator shows that it required skilled attention to many details, was bulky in form, the diaphragms had a habit of cracking occasionally, and there was always some uncertainty concerning the calibration of the diaphragm, since it was loaded statically for this purpose, whereas in use the loads were varying at a very rapid rate ; some hysteresis effect must undoubtedly occur. There was also some doubt concerning the working temperature of the diaphragm and its effect upon the pressure scale.

On the other hand, this indicator yielded excellent diagrams free from friction, vibration, or inertia effects, and at engine speeds up to about 2000 r.p.m. Very small changes in the engine adjustments were readily indicated in the diagrams on the screen.

The Watson-Dalby Indicator.—The original Watson indicator was essentially a research, or laboratory instrument, and so, for commercial purposes, was redesigned by Professor Dalby, in cooperation with its inventor. The commercial instrument [1] is much more compact, stronger in construction, and portable ; the necessary adjustments are also much more convenient.

The Burstall Indicator.—This optical indicator was described at a meeting of the Institution of Mechanical Engineers in January, 1923,[2] and examples of diagrams taken at speeds of 1700 r.p.m. were shown.

The principal features include a pressure piston which transmits its motion to cantilever type beam carrying the pressure mirror, a separate piston displacement mirror, water-cooled indicator

[1] Manufactured by G. Cussons, Ltd., Manchester.
[2] A fuller account is given in the *I.M.E. Proceedings*, January, 1923.

cylinder, and separate water-cooled indicator cock, an inclined all-metal camera box, and an oil supply under pressure to the indicator piston. The reciprocating parts are fairly light, and the whole apparatus compact and portable.

Fig. 208 shows the indicator in sectional plan and elevations. The upper left-hand view shows the water-cooled cylinder, piston, and control spring; the latter is fixed on its right-hand side, and carries the pressure mirror on its left. The piston-rod is hollow, and it is attached to the spring by a spherical ball forming part of the spring. This ball rests in a conical seat in the end of the piston-rod itself, and is fixed in position by a screwed cup in the piston-rod; this form of connection allows freedom of movement without constraint.

The indicator described was designed for a maximum pressure of 600 lb. per square inch, which gave a total piston load of 75 lb. The total motion of the piston, which was about 0·08 inch, and the fact that the upper ball moved in a circle, necessitated a slight barrelling of the piston to prevent stickage; the amount of the barrelling was 0·002 inch. The spring was made of uniform thickness, but of varying width, so as to give equal strength.

The illumination, which is always a difficult problem in high-speed optical indicators, was a Pointolite lamp in this case; it was found necessary to provide adequate ventilation for the bulb. A diaphragm with a square hole of side 0·002 inch was employed as the point source of light.

The indicator spring in this indicator could be calibrated direct, as in the Hopkinson type, by inverting the instrument, and loading with dead weights.

Some excellent diagrams at low and normal speeds up to about 1000 r.p.m. have been obtained, although at the higher speeds there is evidence of vibration waves on the expansion stroke of full-power curves. The indicator in question has been used regularly by students of an engineering college for ordinary engine-testing work.

There is no doubt that the special design, and the forced lubrication of the piston used in this indicator, has extended the useful range over that of the normal types of piston-indicator.

The Midgeley Indicator.—This instrument, which is manufactured by the General Motors Corporation, of Dayton, Ohio, is of the optical type, having a piston for the pressure element. Referring to Fig. 209 (right-hand view) the body of the instrument consists of a steel tube, which screws into one end of the cylinder head, leaving its end as nearly as possible flush with the walls. The piston, which will be seen at the lower end of the sectional diagram referred to, is of the trunk type, of $\frac{9}{16}$ inch diameter, and is carried by an inner sleeve clamped to the body at the upper end and cut spirally so as to form a stiff spring just above the piston.

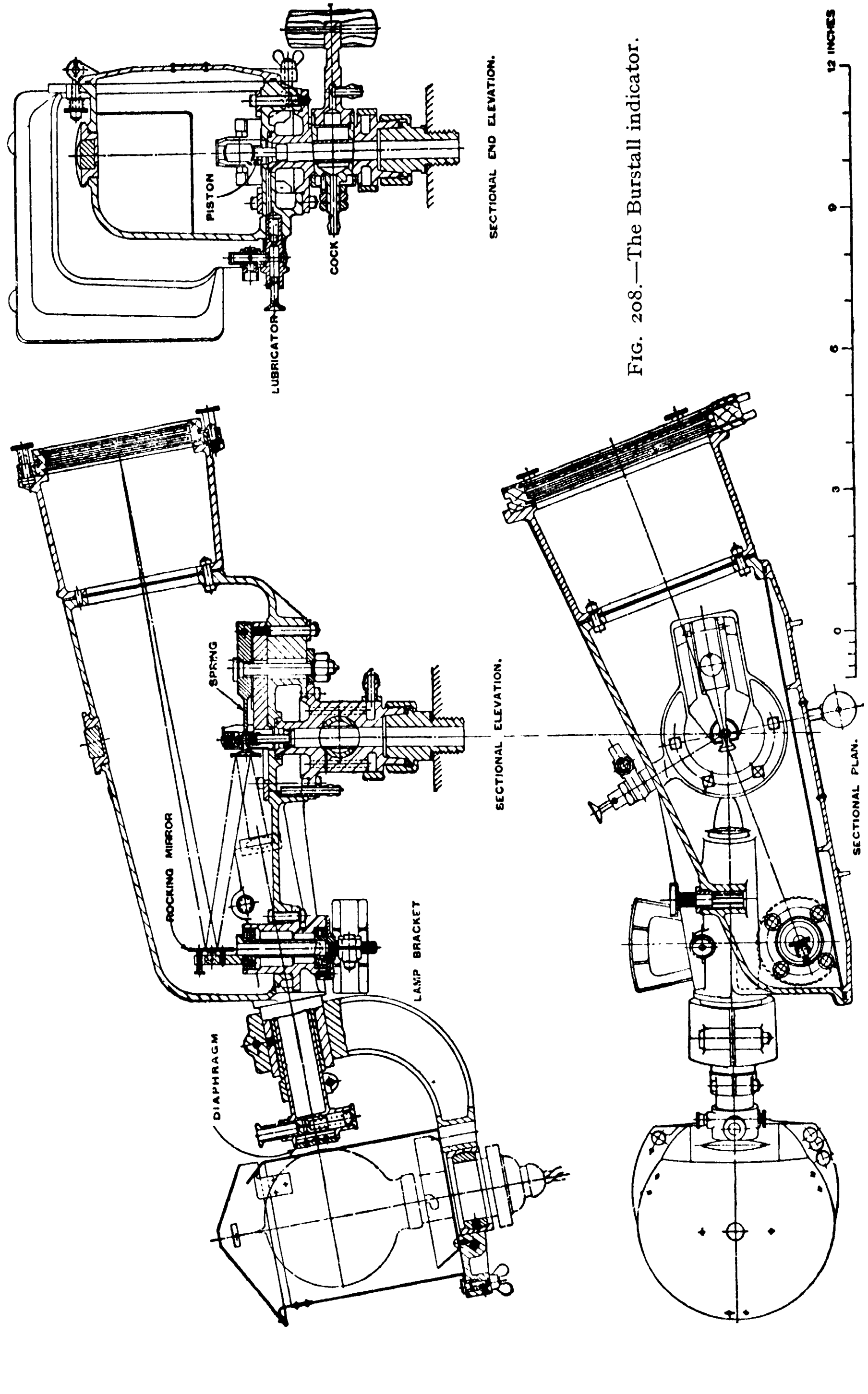


FIG. 208.—The Burstall indicator.

The piston is practically flush with the cylinder walls—an excellent feature as regards absence of " bore waves " and pressure damping,

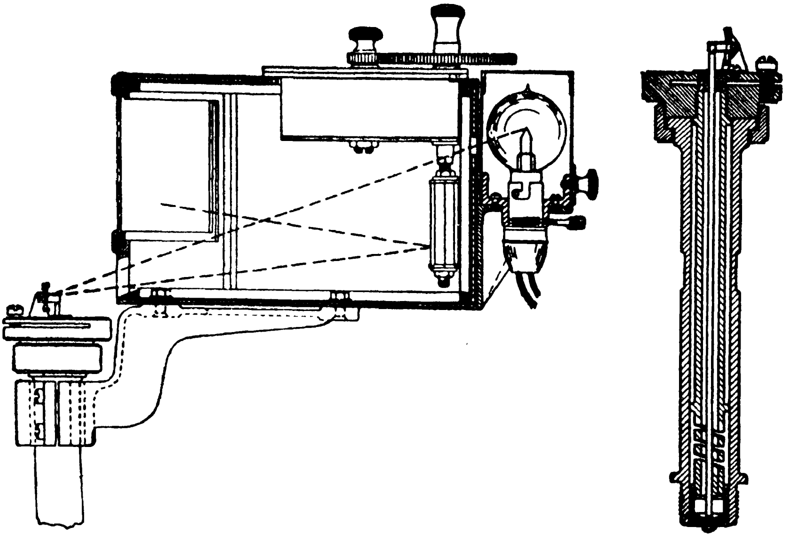


FIG. 209.—The Midgeley indicator, showing pressure element on right.

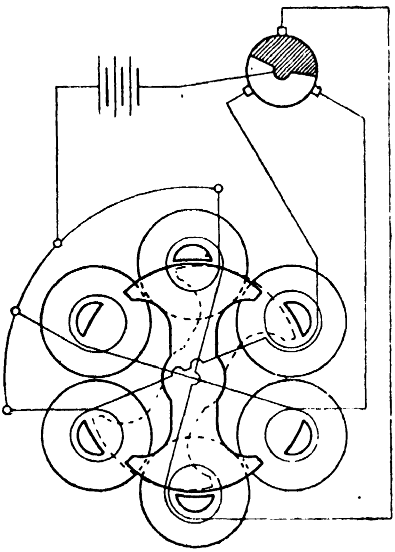


FIG. 210.—Wiring diagram of Midgeley pressure-time synchroniser.

but a detrimental one as regards high temperature effects.

A long rod, or strut, from the piston rocks a pivoted, concave mirror, on a bell-crank lever, at its upper end, proportionally to the pressure amount. A second mirror arrangement for the piston-displacement motion consists of a small, octagonal prism, as shown in the left-hand diagram of Fig. 209, silvered on each face and capable of moving about its axis. When taking ordinary indicator, cards of the pressure-volume type, one face only is used, the prism being oscillated through a small angle by means of a miniature crank and connecting-rod driven at engine speed, the motion being transmitted by means of a fine steel wire or cable to an arm fixed to the mirror spindle. The wire is kept in tension by means of a spring; means are provided for phase adjustment.

The Pressure-time Synchronizer.—The function of the pressure-time synchronizer is to rotate the eight-sided mirror located within the motor box so that its speed may always bear the same definite ratio to the speed of the engine.

The system (Fig. 211) comprises a mechanism for attachment to the engine or dynamometer shaft, and a synchronizer and rotating mirror in the motor box. The synchronism is accomplished electrically by energizing the coils of the synchronizer through a distributor operated by the shaft of the crankshaft attachment.

The distributor is so arranged that during operation the six coils of the synchronizer receive alternate impulses from three

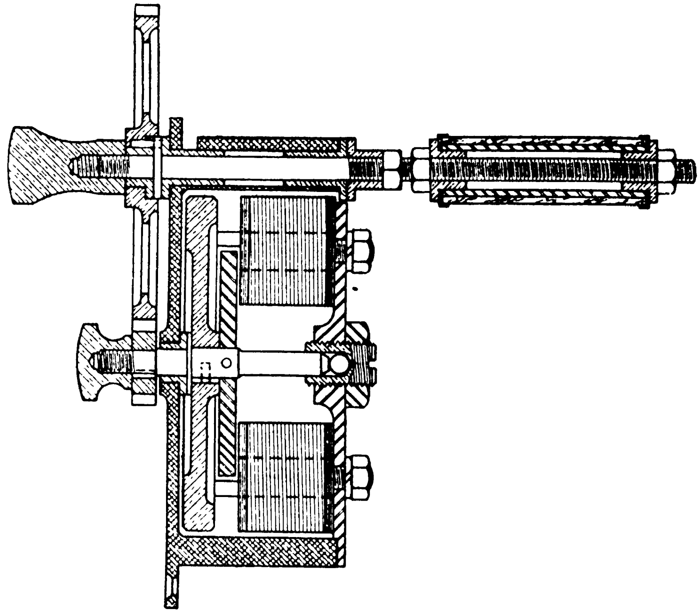


FIG. 211.—The Midgeley pressure-time synchroniser, showing the rotating coils.

contacts in the distributor head. The eight-sided mirror located at the rear of the motor box is mounted on a shaft which is geared to the motor so that it rotates at exactly one-eighth engine speed.

The arc of the curved glass forming the front of the motor box has been made 90° in order that a beam of light focused on and reflected from the rotating mirror is moved across the ground glass once for each two revolutions of the engine. Just at the point when the beam of light reaches the limit of the glass screen, the next succeeding face of the rotating mirror engages the focused beam of light and simultaneously reflects it to the opposite end of the glass screen. In this way, the pressure-time relations within

the engine cylinder for two revolutions, or one cycle, are rendered completely visible on the ground glass as a pressure-time card.

The Pressure-volume Synchronizer.—The function of the pressure-volume synchronizer is to oscillate the eight-sided mirror within the motor box in synchronism with the reciprocating motion of the engine piston (Fig. 212).

Included in the crankshaft attachment is a member driven by a crank attached to the distributor head and having a reciprocating motion in a vertical direction. A fine wire is attached to this member and carried over a light pulley to an arm which transforms the reciprocating motion into an oscillation of the eight-sided mirror about its vertical axis.

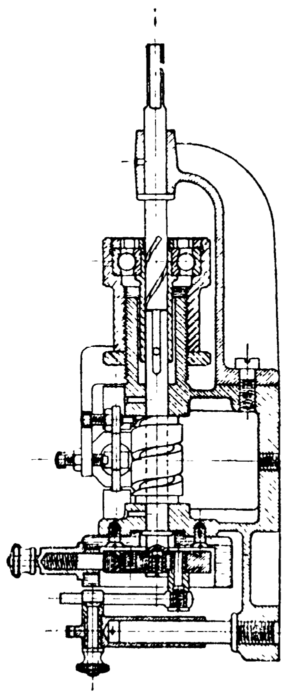


Fig. 212.—The Midgeley pressure-volume synchronizer.

For projecting diagrams, the optical system functions just as in the formation of pressure-time cards, with the exception that the eight-sided mirror is oscillated instead of revolved, only one face of the mirror being used. The mirror oscillates in synchronism with the reciprocation of the engine piston so that the card produced indicates the pressure-volume relations within the cylinder.

The relation between the moving parts is such that the length of the card is just one-half that of the ground glass. An adjustment is provided by means of which the length of the connecting rod for driving the reciprocating member in the crankshaft attachment may be regulated so that it bears the same ratio to its crank as the connecting rod in the engine bears to the engine crank. This eliminates any errors due to connecting rod angularity.

The (Dobbie-McInnes) "Farnboro" Indicator.—This indicator is undoubtedly one of the sturdiest, most portable and accurate of any at present available.

The principle of the pressure element has been given already [1] in the description of the maximum and minimum pressure apparatus, so that it will suffice, here, to describe the recording device, the method of obtaining diagrams, and to refer to some typical examples of the use of this indicator. Fig. 213 illustrates, diagrammatically, the layout of the complete indicator, as applied to one

[1] *Vide* p. 231.

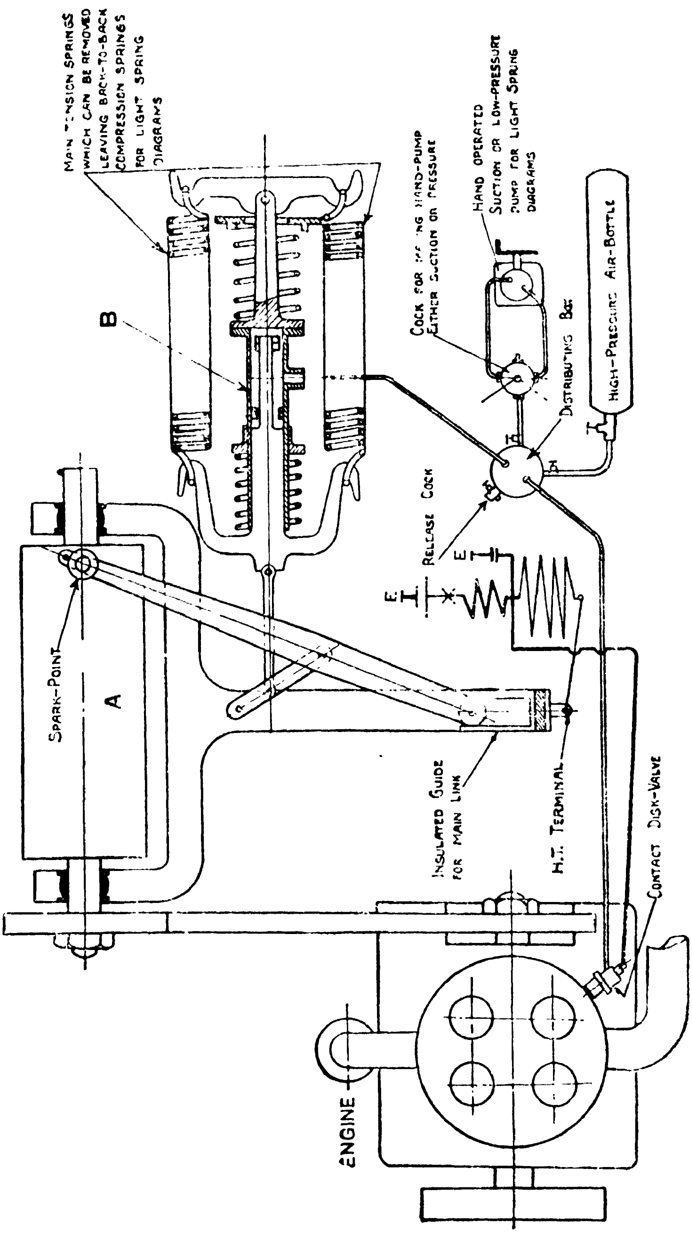


FIG. 213.—Showing arrangement of (Dobbie-McInnes) "Farnboro" indicator.

cylinder of an engine. It has been stated that the principle used depends upon balancing the cylinder pressure with air pressure on the opposite face of a small disc placed very close to the cylinder walls, the point of balance being indicated by the occurrence of a

high-tension spark. For recording purposes, it is only necessary to allow this spark to puncture a special kind of paper wrapped around the surface of a rotating drum, driven off the engine crank, or timing shaft, the result being a pressure diagram built up of a large number of fine punctures, on time (or crank-angle) base.

A more recent improvement in recording is the introduction of a method whereby black diagrams are obtained from the spark on the white surface of the paper, thus enabling immediate examination of the diagram to be made. This result is obtained by coating the recording drum with a special preparation known as " Lectrene " which is in the form of a solution and is applied with a brush.

Referring to Fig. 213, it will be seen that the air-pressure supply is connected to a distributing box, whence it goes (*a*) to the balance valve unit on the cylinder, and (*b*) to a cylinder B, containing a spring-constrained piston. The latter is connected by means of the linkage shown to a lever, the end of which is provided with a sparking point ; the motion of this point is so arranged that it is a rectilinear one parallel to the axis of the recording drum A. Thus the distance of the sparking point from a datum line on the right-hand side of the drum A, is proportional to the pressure in the cylinder B, and therefore—since this is in direct communication with the contact disc, or balance valve—to the cylinder pressure, when sparking occurs at the point over the drum.

The drum A is earth connected, so that the high-tension spark from the end of the insulated link passes through the paper wrapped on the drum every time the air pressure balances the cylinder pressure. The drum A is provided with a chain sprocket, driven by a chain from a similar one on the engine crank—or timing shaft. A simple form of dog clutch is arranged on the left-hand side of the drum (Fig. 213) to throw it into engagement with the chain drive, so that the drum is only in use when required. The drum is speeded up by special means to approximately the engine drive speed before the clutch is put home. A hand-operated brake is also provided on the drum, for the purpose of eliminating backlash when taking diagrams.

The circumference of the drum is 14·4 inches, and its length 9·5 inches ; the diagrams obtained are thus quite large ones, measuring 14·4 inches by about 7½ inches.

Method of Building up Diagram.—This indicator gives an average pressure-time diagram, consisting of a number of dots or perforations, each pair of which belongs to a different cycle.

Referring to Fig. 215, it will be seen that as the air pressure is increased gradually from, say, atmospheric value, it will equal the cylinder pressure at a point *a*, on the compression stroke, and then at another point *b* on the expansion stroke. Similarly, there will

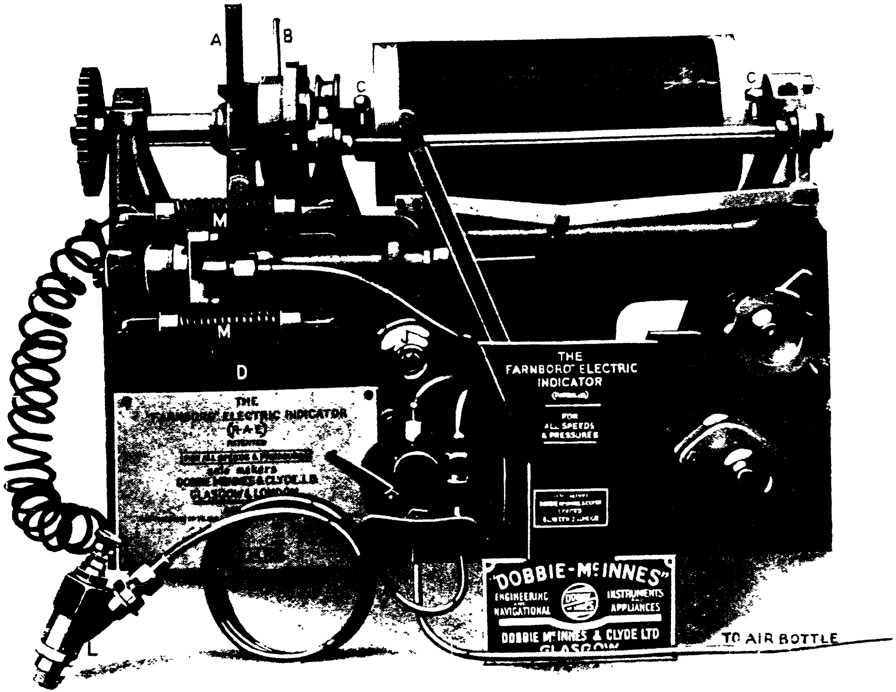


FIG. 214. The (Dobbie-McInnes) "Farnboro" indicator.

To face page 260.

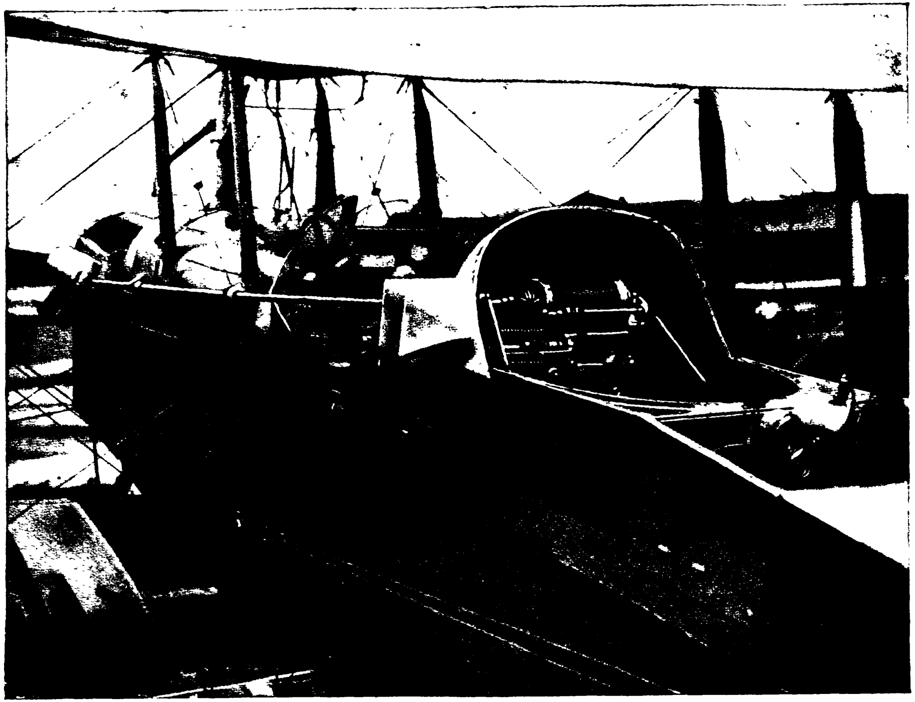


FIG. 220.—Showing the "Farnboro" indicator fitted up for aeroplane flight tests.

[To face page 261.

be two higher pressure points of balance, *c* and *d*, on the compression and expansion strokes of the next cycle. In this way an average diagram is built up, consisting of pairs of points on opposite sides, contributed by each individual cycle. The course of the spark is, as it were, a zig-zag one *abcdefgh* on the paper development.

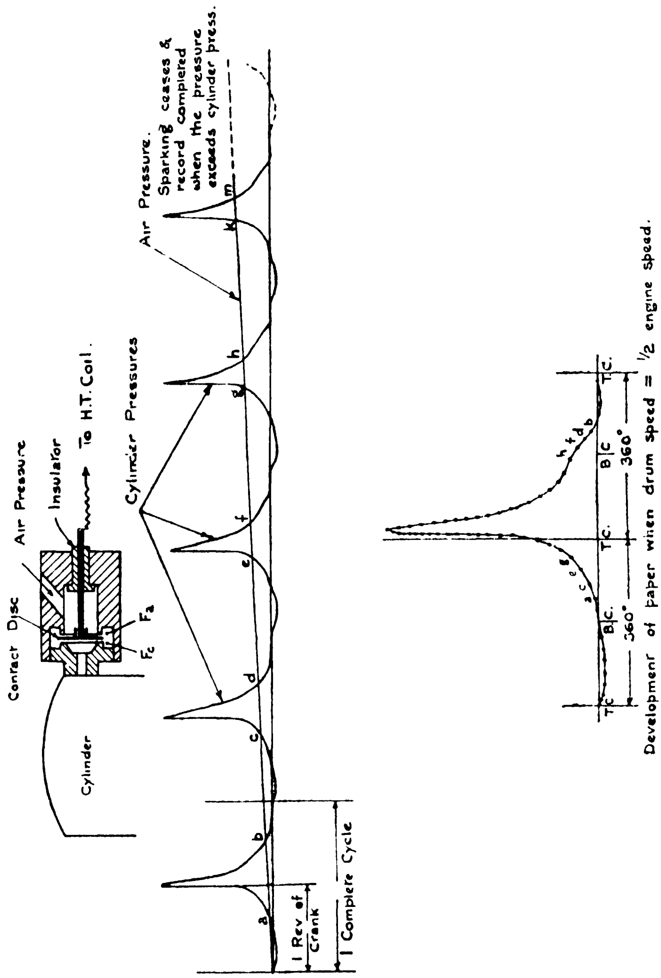


FIG. 215.—Showing method of building up the pressure diagram.

Fig. 216 is a reproduction of an actual diagram, but with an average line drawn in by hand. It was obtained from a six-cylinder water-cooled aircraft engine of 180 mm. bore and stroke, with a compression ratio of 5·59. The maximum explosion pressure obtained was 630 lb. square inch at 1400 r.p.m.

The letters TC and BC refer to the top and bottom dead-centres respectively.

Fig. 217 shows the pressure-volume diagram which was obtained by a graphical conversion process from the above.

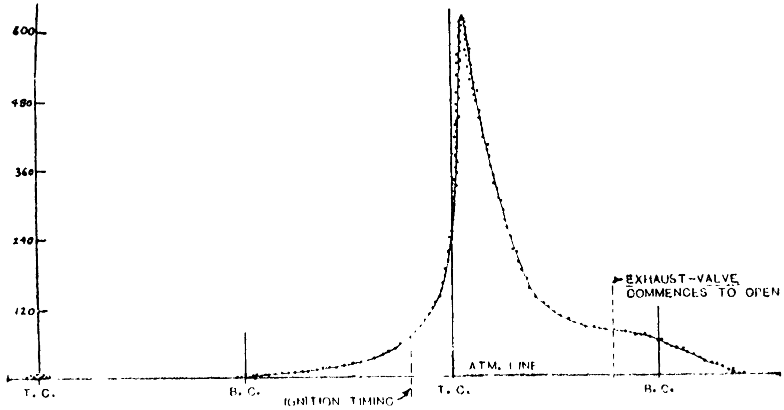


FIG. 216.

Fig. 218 shows three superposed diagrams taken from an aeroplane engine in flight at altitudes of 1000, 10,000 and 21,000 feet, for the diagrams 4, 5, and 6 respectively. The corresponding air-speeds were 91, 84 and 73 m.p.h., and engine-speeds 2260, 2080 and 1890 r.p.m.

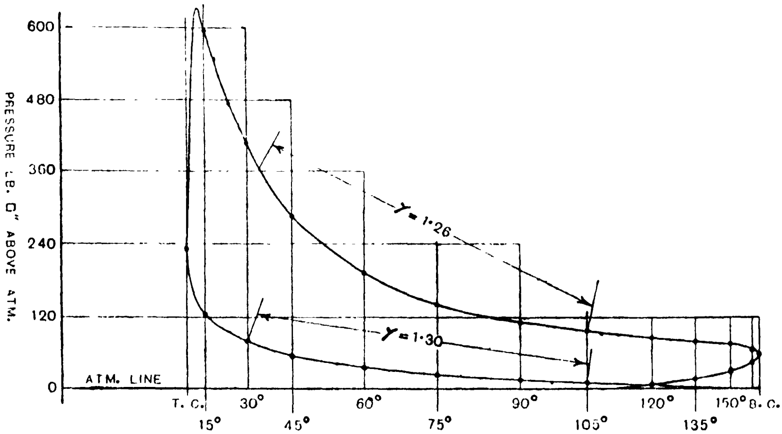


FIG. 217.

The diagrams shown are about one-half the actual size, it having been necessary to reduce the scale for reproduction purposes. The pressure scale corresponding to the reduced diagrams is 1 inch = 75 lb. sq. in. The time scale is 1 inch $\pm$ 64·25°.

The corresponding pressure-volume curves to the diagrams shown in Fig. 218 (which are on a time-base) are reproduced in Fig. 219. This illustration also gives some interesting data relating to the engine speed, mean effective pressure and indicated h.p. at the three heights mentioned.

The pressure-volume diagrams shown have also been reduced to about one-half the original size, in reproduction.

Pressure Indicating Piston.—The air-pressure piston is connected to the distribution box (see Fig. 213) by a length of tubing

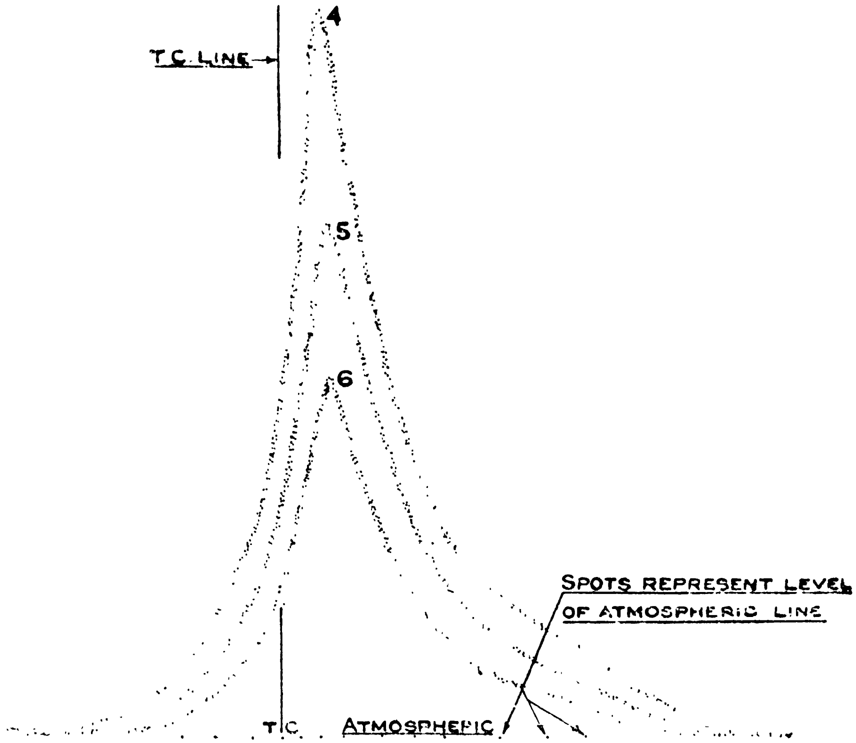
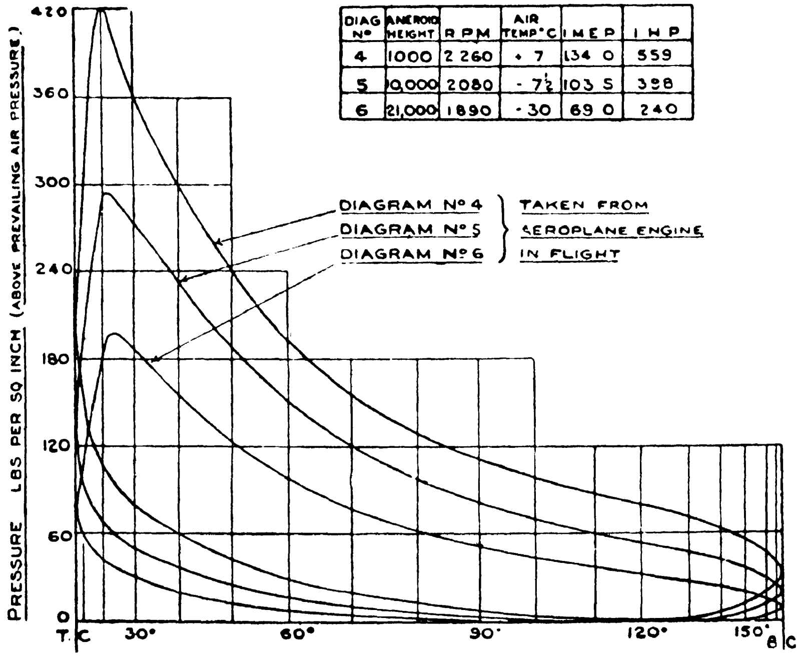


FIG. 218.—Indicator diagrams taken from aeroplane engine in flight.

equal to that between the engine and the distributing box, to avoid pressure differences due to different lengths. The piston-rod has a stirrup attached to its outer end, to which two calibrated tension springs are attached. Another pair of light-compression springs, placed back to back, is also provided for the purpose of enabling "light spring" diagrams to be obtained. These springs, being of equal strength and load rate, give an atmospheric line at a point midway in the piston travel. Suctions as low as 12 lb. per square inch, and pressures up to about 20 lb. per square inch can readily be obtained.

There is an auxiliary hand-operated suction, or low-pressure pump, which is used instead of the high-pressure air supply for obtaining these "light spring" diagrams. Means are also provided for obtaining the usual atmospheric and top dead-centre lines.

The Contact Disc Valve Unit.—Fig. 221 illustrates the improved Type A contact disc valve unit. This is provided with a short water-cooled cock and a fixed central electrode insulated with mica. The disc valve is cooled and cut off from the engine cylinder gases by the cock when the indicator is not in use. The electrode and its guide are made in one piece so as to minimize any risk of air leakage.



DIAG Nº	ANEROID HEIGHT	R P M	AIR TEMP °C	I M E P	I H P
4	1000	2 260	+ 7	134 0	559
5	10,000	2080	- 7½	103 5	398
6	21,000	1890	- 30	69 0	240

FIG. 219.—Pressure-volume diagrams corresponding to those shown in Fig. 218.

The travel of the disc valve is an important factor in the satisfactory operation of this indicator. If the disc travel is too great there will be a noticeable "missing" at the top of the expansion line. The travel allowed to the disc valve is ·006-·010 inch, the smaller travel being preferable.

Combined Sparking-plug Pressure Unit.—The unit shown in Fig. 222 enables the pressure element of the "Farnboro" indicator to be combined with the sparking plug so that it can be screwed into the ordinary cylinder head sparking plug. A standard mica type plug insulator is used for the latter, whilst another insulator re-machined in employed for the insulator of the valve. The valve

is $\frac{5}{16}$ inch in diameter having a raised rim on each side 0·007 inch high and 0·007 inch wide. It seats on Stellite washers with a copper spacer allowing about 0·005 inch movement. The balancing pressure is led to the space about the valve through a steel tube brazed to the body and through holes in the spacer beneath the insulator The valve stem guide is vented to eliminate any pumping action when oil is tapped above the valve stem. The unit in question was developed by the American Continental Motors Corporation Research Laboratory.

Conversion of Diagrams.—Each diagram obtained consists of a black-backed paper sheet perforated with a large number of fine holes, each of which appears as a black dot on the white side

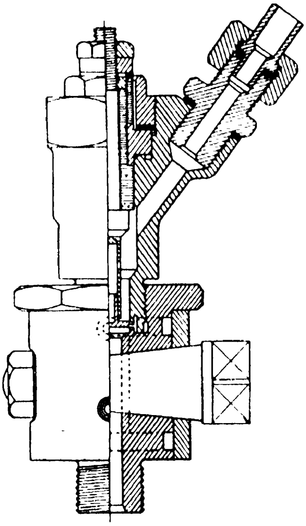


FIG. 221.—Contact disc valve unit.

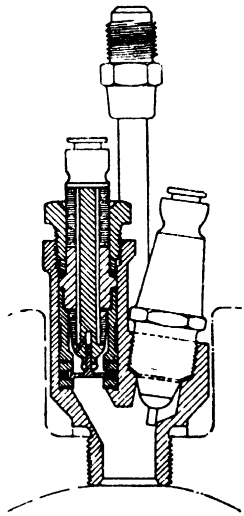


FIG. 222.—Combined sparking-plug pressure unit.

of the sheet. If desired, the average diagram may be drawn with a pencil through the dots, and either these or the perforations of the paper may be used if photographed duplicates are required.

This pressure crank-angle diagram can readily be converted into a pressure-volume diagram by using a transparent scale, graduated in degrees, and corresponding to the connecting-rod crank ratio of the particular engine.

General Particulars.—The indicator described has been used successfully at engine speeds up to about 6000 r.p.m., and some interesting diagrams have been obtained from aircraft engines in flight at different altitudes up to 21,000 feet, which have yielded useful information on the altitude-power relation.

The indicator recorder can be placed well away from the engine

itself,[1] so that it is well adapted for the road testing of automobiles, and for aircraft engine testing, in flight, etc.

Multi-cylinder engines can be indicated very conveniently, since only the balance-valve unit is required on each cylinder and a length of air-pressure tubing to a distributor cock. The one indicating drum does for any number of cylinders; four cylinders can thus be indicated, one after the other, the resulting diagrams being in their correct sequence.

In some cases a multiple change-over switch may be advisable.

The average time required to obtain a complete diagram of the pressure-time type is about 15 seconds; the time to convert this into a p.v. diagram is about three minutes on the average. In particular the apparatus is very suitable for studying rates of ignition, or explosion, since the ordinates are well separated on the explosion curve. For fuel injection engines the pressure-time curves obtained are particularly useful. The diagrams obtained are sufficiently large (14·4 by 7½ inches) to enable accurate measurements to be made therefrom.

Diesel Engine Diagrams.—The "Farnboro" indicator has been somewhat extensively used in connection with Diesel engine research. With engines operating on this cycle it is possible to measure, from the time-base indicator diagram, the change in the rate of pressure rise due to the *cooling effect of the injected charge*, the *time-interval between injection and commencement of burning* and the *rate of pressure rise during burning*.

For this purpose a special differential plunger-valve unit is available. For Diesel engine fuel-injection pressure measurements this special unit may be inserted anywhere in the fuel delivery pipe without interrupting the flow of oil. It provides a diagram of injector pressure variations on the same card as the ordinary cylinder pressure diagram and simultaneously with it.

This is brought about by operating the disc valve by means of two opposed plungers, that on the air side having ten times the area of the plunger on the fuel side; the fuel diagram has therefore one-tenth of the cylinder diagram pressure scale. This unit has successfully been used to measure cylinder pressures greatly in excess of 1125 lb. per square inch.

The differential plunger unit is shown in Fig. 223; and combined diagrams obtained with this unit and the normal cylinder pressure one are shown in Fig. 224. The plunger unit is screwed at the bottom to a Tee-piece inserted in the fuel line.

High Pressure Gas Diagrams.—For *high pressure gas diagrams* a modified type of differential plunger unit having a 4 to 1 ratio and fitted with a shut-off cock and water-cooling connection

[1] It has given satisfactory results at distances up to 12 feet away from the engine.

is now available. It has been used successfully to indicate a special aircraft engine under supercharged conditions to give a B.M.E.P. of over 250 lb. per square inch ; it has been used also for still higher pressures in gas compressor tests. The maximum pressure with the ordinary disc valve unit for the strongest spring R is 1125 lb. per square inch. For the 10 : 1 differential plunger unit this value is increased to 11,250 lb. per square inch, and for the 4 : 1 plunger, 4500 lb. per square inch. Other springs can be supplied for lower pressures down to about one-quarter of the values mentioned.

Top Dead-centre Line.—A later improvement to the "Farnboro" indicator consists in sparking the T.D.C. line on to the diagram paper whilst indicating. This is accomplished by another circuit consisting of a contact breaker fitted to the engine crankshaft, an induction coil, 6-volt battery and switch. By

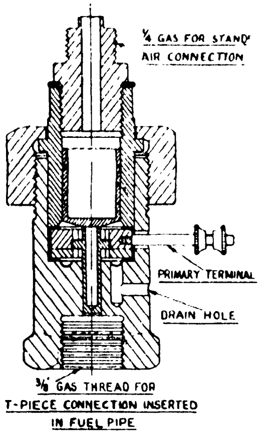


FIG. 223.—Differential plunger unit for Diesel engine tests.

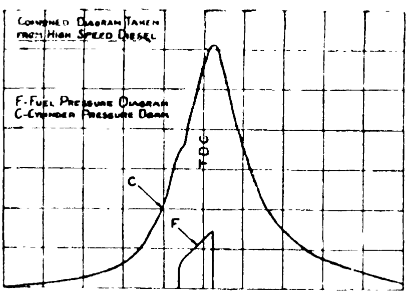


FIG. 224.—Examples of pressure diagrams from Diesel engines.

allowing the secondary spark to puncture for test purposes a slip of paper gummed to the engine flywheel at T.D.C., accurate setting of the contact breaker and phasing of the diagram are assured.

Pressure Calibration Lines.—Another improvement is obtained by introducing a further circuit and a test pressure gauge such that constant pressure lines may be sparked on the diagram at predetermined intervals. This circuit consists of a vibrator and push-switch connected to the same auxiliary coil and battery referred to above. At each pressure increment of, say, 10 or 100 lb. per square inch or 1 or 10 kg. per cm.^{2} as indicated on the gauge, the switch is depressed and a stream of sparks records a horizontal line across the diagram sheet.

Fig. 226 shows a petrol engine diagram taken with the two auxiliary circuits in operation.

Spray Valve Lift Unit.—A recent development in connection with the "Farnboro" indicator is that of a unit for obtaining diagrams of fuel spray valve lift on the same record paper as those of cylinder pressure and fuel injection pressure, the T.D.C. and crank angle scales being common to all the diagrams.

The spray valve lift unit is mounted at the top of the fuel injector and has a pair of electrical contacts, one above and the other

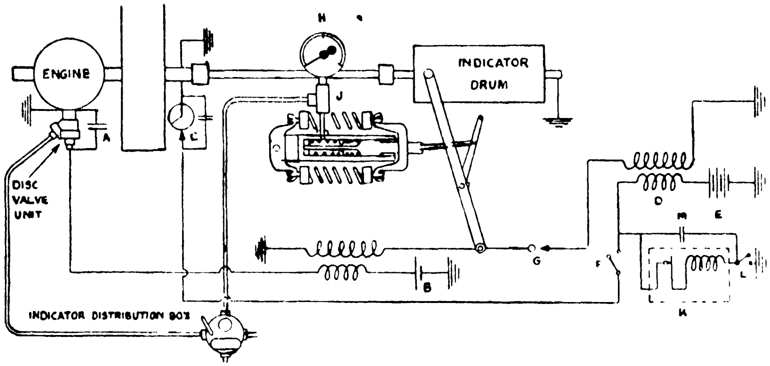


FIG. 225.—Diagrammatic sketch of "Farnboro" indicator with auxiliary circuits.

A. Additional condenser across D.V. unit. B. 2- or 4-v. battery. C. Contact breaker with additional condenser. D. Induction coil. E. 6-v. battery. F. Tumbler switch. G. $\frac{1}{4}$-in. spark gap acting as non-return valve. H. Test pressure gauge. J. T-piece between gauge and oil reservoir. K. Vibrator. L. Spring- or push-switch. M. Additional condenser across vibrator.

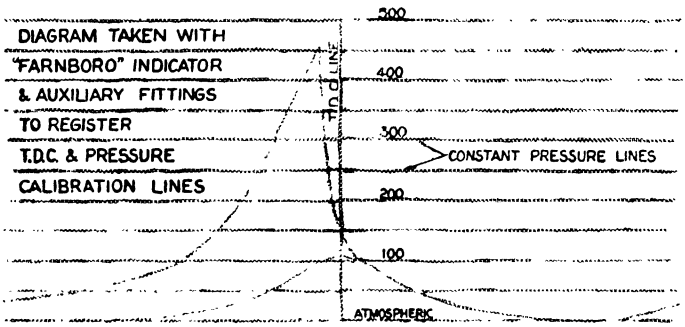


FIG. 226.

below a fitting on the top of the needle valve. Each time an electric contact is made a spark occurs on the recording drum. A micrometer adjusting screw with a scale graduated to read in thousandths of an inch enables the various positions of the contacts to be adjusted, corresponding with the fuel valve positions. Both up and down positions of the fuel valve can be obtained so as to give a complete record on the drum.

Bureau of Standards Balanced Diaphragm Indicator.—A balanced diaphragm type indicator is described in Report No. 107 of the American Advisory Committee for Aeronautics, which has been used successfully at speeds up to 2600 r.p.m., and from minus 10 to plus 1000 lb. per square inch; the results obtained are stated to be as accurate as those from a standard 6-inch pressure gauge. The pressure element, which is illustrated in Fig. 227, consists of a thin metallic diaphragm 1, dividing the chamber into two parts, the lower of which is in direct communication with the engine cylinder at 2, a water-cooled chamber 3 being provided, whilst the upper side 4 communicates with a compressed air, or gas supply. The disc has a limiting motion of 0·13 mm. (0·005 in.). Above and below it are circular plates, about 5 mm. thick, perforated with No. 60 drill holes, and surfaced with corrugations arranged concentrically, the object being to prevent distortion of the diaphragm beyond its elastic limit.

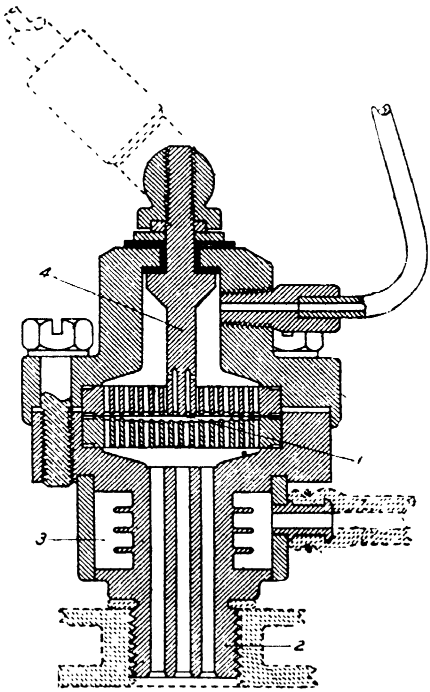


FIG. 227.—Balanced diaphragm indicator.

The diaphragm employed is made either of phosphor bronze or steel, and measures 30 mm. (about $1\frac{1}{4}$ in.) diameter, and about 0·08 to 0·13 mm. (0·003 to 0·006 in.) thick. The time-lag of this diaphragm can be calculated, and the results show it to be unimportant. In order to indicate the points of balance in the cycle, between the cylinder and air pressures on the diaphragm, the centre part 4 of the upper support is insulated, electrically, from the body. This part forms an electrode, which is connected in series with a telephone as a detector and one side of a battery, of which the other pole is earthed on the engine frame, and therefore the diaphragm. When the diaphragm moves upward against the electrode it closes the circuit, clicking the telephone, and thus indicating that the cylinder pressure is greater. As in the previously described indicator, there are two such balancing points to every single cycle.

In order to obtain a crank-position base for these balancing pressures, or in other words to locate the pressure values indicated, an electric timer is employed. This consists of a rotating disc,

affixed to the crank or cam shaft. The body of the disc is insulated, but in the periphery is inlaid a narrow strip of brass subtending an angle of one-half degree. This serves as an electric contact in the telephone circuit, by its rotation past a " fixed " brush. The position of this brush can be varied relatively to the crankshaft of the engine, so that electrical contact between it and the half-degree strip can be established at any part of the crank circle. In this way the air pressure balancing the cylinder pressure can be located for all parts of the cycle, since the position of the " rotating " brush is known. The pressure corresponding to each position of the brush (and, therefore, to each known crank-angle) is measured on a standard pressure gauge of the 15 to 100 lb. type for pressures of 0 to 100 lb. square inch, and on a 100 to 1000 lb. type gauge for the higher pressures. For lower pressures, such as those corresponding to the suction and exhaust strokes, a manometer is employed. The arrangement of the apparatus is shown diagrammatically in Fig. 228.

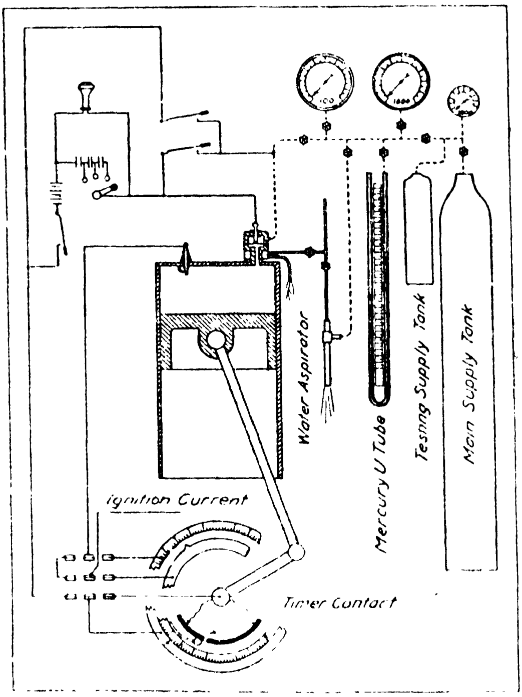


FIG. 228.—Schematic diagram of indicator assembly. Showing the pressure element, timer pressure system and electrical circuits in their relation to an engine cylinder.

This instrument can also be used as a maximum and minimum pressure-measurement device.

It should be mentioned that the point of balance of the two pressures on the diaphragm is indicated by a definite change in action of the telephone, from clicking every engine cycle to complete silence. When the successive cycles do not repeat, the abrupt change from clicks to silence is replaced by a range of pressure over which the clicking becomes irregular as the gauge pressure is raised, the clicks ceasing entirely at a pressure equal to that of the highest cylinder pressure reached.

The operator notes the gauge pressure and the timer position at the balance points and records them. It is thus obvious that

this method will give an ultimate result which represents the average of a very large number of consecutive cycles, so that uniformity of engine running is essential.

Optical Balanced Pressure Indicator.—A balanced pressure indicator employing a somewhat similar pressure unit to the "Farnboro" one, but with an optical method of recording the diagrams has been developed by R. Brandt and H. Viehmann of the German Institute for Aeronautical Research.[1]

It uses the balanced disc and compressed air method, but above the disc there is an iron rod insulated from the body of the pressure plug. When the cylinder pressure just overcomes the applied air pressure the disc lifts and makes contact with the lower end of the

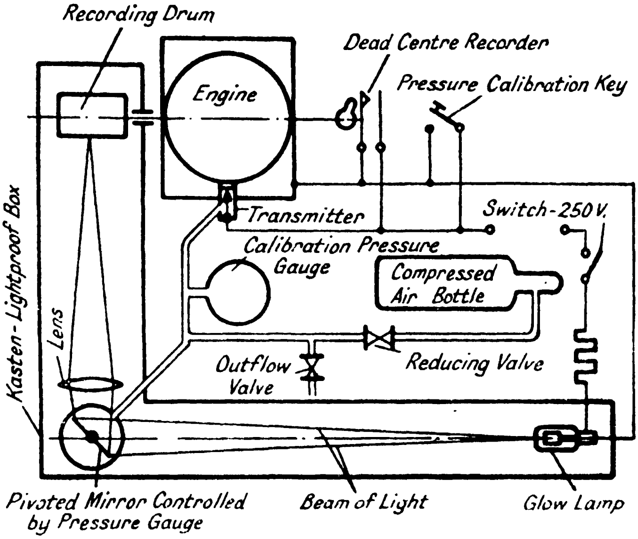


FIG. 229.—Brandt and Viehmann optical balanced pressure indicator.

iron rod, thus completing an electrical circuit which includes a source of current and a glow lamp; the latter lights during the period that the cylinder pressure exceeds the air pressure. The optical arrangement for transmitting the pressure values to the recording drum is shown, schematically, in Fig. 229. The high intensity glow lamp throws a beam of light on to a pivoted mirror, the angular deflections of which are controlled by a pressure gauge in the compressed air supply to the cylinder pressure unit. This is arranged to give linear deflections which are proportional to pressures. The beam of light is reflected from the mirror and focused by a lens so as to form a spot of light on the photographic paper wrapped around the drum; the latter is rotated at engine speed.

[1] *Autom. Industries*, Sept. 22, 1934.

It will be seen that a pressure line, normal to the crank-angle line, is drawn during each revolution of the crankshaft, the distance from the crank-angle line being proportional to the air pressure and the length shows during which portion of the stroke the pressure in the cylinder is greater than the air pressure. If the latter is slowly increased whilst the engine is working, the indicator produces a plane surface diagram composed of straight lines whose outline or contour represent the mean of a number of consecutive pressure-time diagrams. A pressure scale can be inscribed on this diagram by means of the pressure-calibrating button shown in Fig. 229, the pressure being simultaneously read off an accurately calibrated pressure gauge (not shown in Fig. 229).

The dead-centre position is recorded on the diagram by means

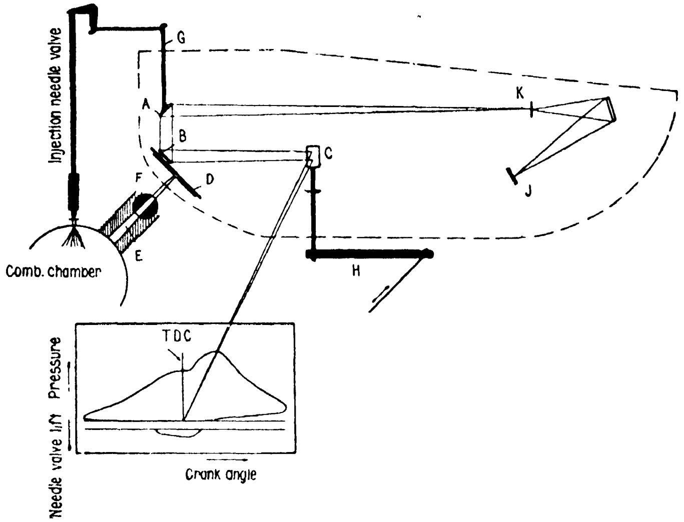


FIG. 230.—Arrangement of the B.P.C. optical indicator.

of a cam mounted on the end of the crankshaft; this closes an electric contact momentarily at the T.D.C. position. In this way a series of points is marked on the diagram and these all lie on a straight line normal to the crank-angle zero line.

The apparatus described is used also for obtaining single *maximum cylinder pressure values*, for inlet and exhaust stroke diagrams and pressure diagrams corresponding to two different points in the combustion chamber, e.g. one in the cylinder head and the other in the ante-chamber of a compression-ignition engine.

The diaphragm used has a frequency of 100,000 cycles per sec., i.e. well above any engine vibrations or inertia effects. The glow lamp belongs to the neon class as it must evidently be free from lag effects in illumination. The motion of the deflecting mirror is so slow, relatively, that its inertia effects are negligible. The

diaphragm of the pressure unit interrupts a current of 10 to 20 milliamperes at 220 volts.

The B.P.C. Indicator.—The indicator shown schematically in Fig. 230 was produced by the research laboratory of the Batavian Petroleum Company, Holland, for use on high speed internal combustion engines, e.g. compression-ignition and petrol engines, and it claims to have solved successfully the problem by using a diaphragm as the pressure element, mounting directly to this a prism in order to dispense with any other transmitting mechanism ; the principle employed is basically the same as in the original indicator of Prof. J. Perry,[1] in which a mirror was mounted direct on to the pressure diaphragm.

The indicator employs a reciprocating movement to serve as a scale for the crank position and time for at least 60° of the diagram and, after calibration, even up to 120°. The advantage is a continuous diagram and a more reliable drive, as the rotating movement is accurate only if connected direct to the crankshaft. On the other hand, the reciprocating movement can be taken from an eccentric driven by the crankshaft by means of a simple steel tape. When indicating optically the inertia forces can be kept small, as only a small mirror rotates around its axis ; moreover, the stroke of the movement can be small, which secures a compact drive. It should be remembered, however, that a stroke of say 10 mm. of the eccentric is at least ten times enlarged on the diagram, so that deviations of 0·1 mm. due to vibration are clearly visible. The guiding of the transmission tape between eccentric and indicator, therefore, is of importance ; specially constructed guide rollers must be made for this purpose.

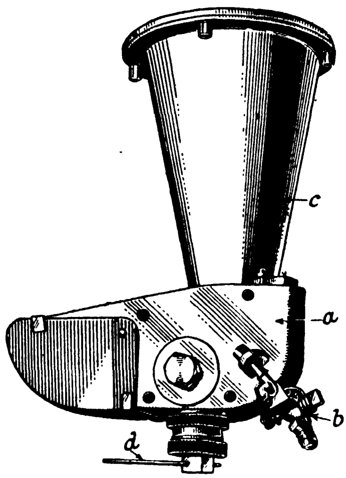


FIG. 231.—The B.P.C. optical indicator.

Fig. 231 shows the indicator as it is mounted to the cylinder. It consists of a solid block of bronze *a* in which the pressure element and the optical parts are fitted free from vibration. The block is connected to the engine by means of the cock *b* and fastened with stay bolts. The funnel *c* forms the camera which is equipped with a ground-glass slide (for observation purpose) and with a carrier for photographic plates or films of 9 × 12 cm. The funnel

[1] *Proc. Phys. Soc.*, **11**, 1891 (151).

is adjustable over 180° so as to place the image always in a favourable position. The lever *d* receives a reciprocating movement from the indicator drive and is intended to obtain the abscissa of the diagram. This lever can be fastened in various positions.

Fig. 230 shows the path of the rays of light and the principal parts. The light emerging from the lamp J (4-6 volts) is condensed on the diaphragm K, a sharp-pointed image of which is formed on the photographic plate. The light falls first on a mirror A (which will be discussed later) and then on to the mirror B fixed on the diaphragm D. It is then reflected on to the abscissa mirror C which projects it into the camera. (In practice, instead of mirrors, use

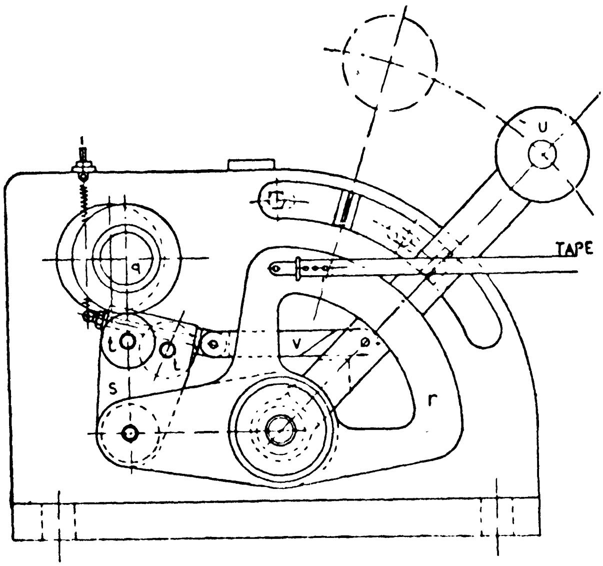


FIG. 232.—Driving block of the B.P.C. indicator.

has been made of totally reflecting prisms in order to avoid corrosion.) The pressure element D is a steel plate of an effective diameter of 28 mm. while the pressure chamber is 0·1 mm. deep only. The pressure chamber is connected to the combustion space by means of the tube E carrying the cock F. The total volume of connecting tube and pressure chamber is less than 0·3 c.c., of which about 60 cu. mm. is occupied by the pressure chamber; thus neither the cock nor the diaphragm are heated.

The indicator can be fixed closed to the engine under test. The mirror A is fixed on a lever G serving for the adjustment of the height of the diagram as well as for recording special movements such as those of *fuel injection needle valves* (as shown on the left in Fig. 230).

The drive consists of a driving block, from which a steel tape is

operated, and the rollers guiding this tape to the lever H (Fig. 230). The block is fixed to the engine so that the driving shaft can be connected firmly to the crankshaft of the engine. The guide rollers are connected with the case of the indicator and with the driving block by means of a suitable frame. This frame must be made according to circumstances.

The driving block (Fig. 232) contains a shaft q with an eccentric and a concentric disc of equal size. The tape is connected to the fulcrum lever r which is driven by the shaft q *via* a connecting piece s carrying two rollers t. By means of a handle u, connected to s by a rod v, one of the rollers t can be made to run on the eccentric or the concentric disc. When the roller runs on the eccentric disc, the spot of light produces a diagram of a certain length. When the other roller runs on the concentric disc the tape does not move, and the spot of light stops its horizontal movement in the middle of the length of the diagram. The driving shaft q should be coupled to the engine shaft so that the spot where the light thus comes to rest corresponds, e.g. to the top dead-centre in the diagram. With running engine and the pressure element working, this dead-centre can therefore always be recorded then as a vertical line on the diagram (see Fig. 230). The length of the diagram can thus be adjusted by means of the lever H (Fig. 230) so that a useful scale for time or positions of crank is produced, even when the ends of the diagram do not fall on the plate, or, in the case of photographic records when the ends of the diagram are indefinite owing to over exposure.

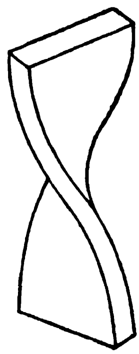


FIG. 233.—Twisted tubular pressure element of Bauer indicator.

The Bauer Indicator.—An optical indicator, devised by Dr. S. G. Bauer [1] for the purpose of giving accurate absolute values of pressures in combustion chambers and separate records of individual cycles, utilizes a twisted alloy steel tubular pressure element of the type shown in Fig. 233. Such a tube possesses the property of "untwisting" under increasing internal pressure. One end of this tube communicates with the combustion chamber and the other end is closed by a brazed-in stainless steel plug, the outside face of the latter being ground flat and polished to form a mirror. The dimensions of the pressure element were such that, measured at a distance of 1 metre, 1 mm. displacement of the spot of light corresponded to a pressure increase of 17·65 lb. per square inch. The natural frequency was 8200 cycles per second. It was found that there were no hysteresis effects with this element. Its

[1] Fully described in *The Engineer*, Aug. 19, 1938.

deflections were accurately proportional to the pressure for small deflections, but large deflections were not allowable if the stresses were to be kept within the fatigue limit of the steel. It will be noted that the pressure variations are measured by the angular movements of the twisted tube about its longitudinal axis; in this connection it was found that no torsional movements of the pressure element could be produced by engine movements or vibrations—this is a marked advantage of the arrangement in question. The layout of the optical units is shown in Fig. 234 (lower part). The light from the filament I of a 12-volt bulb is focused through the condenser lens C_1 on to a vertical slot S_1, which, in turn, serves as a new source of light as a sharp vertical line. A full-scale image of the latter is formed on the photographic film around the drum F with the aid of the lens A and the flat mirror E on the end of the

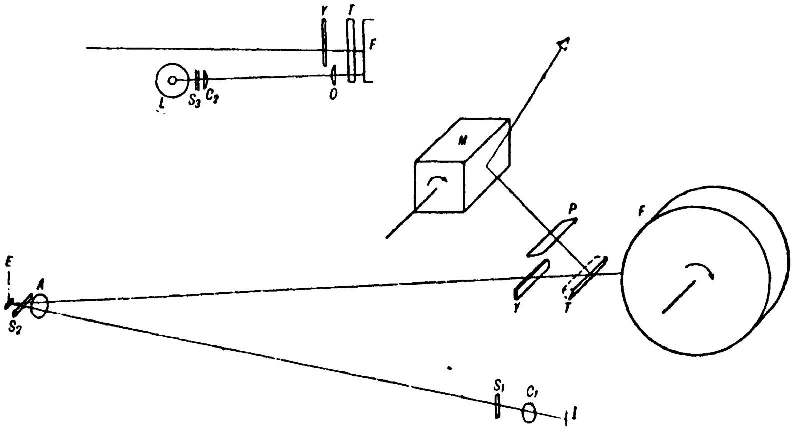


FIG. 234.—Optical arrangement of the Bauer indicator.

indicator element; if the mirror E swivels round the vertical axis of the element under the influence of a change in internal pressure, the image of S_1 moves across the film in proportion to the change of pressure. A transverse movement of the mirror has no effect on the image on the film, but an angular movement round a horizontal axis perpendicular to the path of light would cause a vertical displacement of this image. A horizontal slot S_2 is therefore placed in the path of light, right in front of the mirror E. An image of S_2 is formed in a vertical plane on the film by the cylindrical lens Y. In a horizontal plane this lens has no influence on the image of S_1 on the film. The consequence is that all the light which reaches the film is concentrated in the vertical dimension to the width of the slot S_2. A very small square of light, virtually a spot, is thereby obtained on the film which can be displaced only horizontally by the angular deflection of E round a vertical axis. No

engine vibration has any influence on the spot of light, as both the slots, by the images of which it is formed, are rigidly connected to the camera.

The film is held on a 6-inch drum F, which revolves at a suitable speed, so that a pressure-time diagram is obtained. The shutter T carries a mirror at such an angle that when the shutter is down the light is deflected on to a semi-transparent screen P which is viewed through a set of rotating mirrors M. This draws the periodical pressure deflections out into a time base diagram.

If a record of a diagram is desired, the shutter is lifted by an electro-magnet, and the lamp is over-loaded to give intense illum-

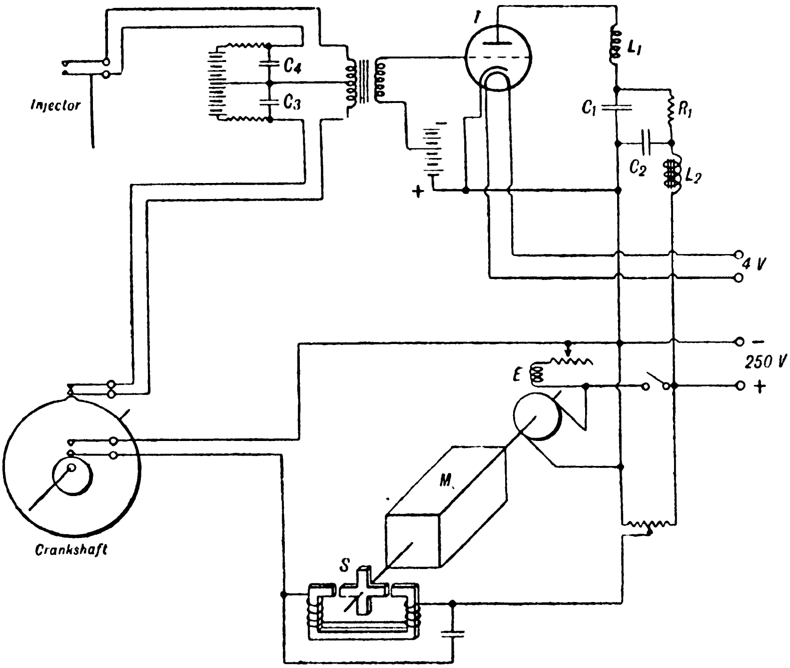


FIG. 235.—Complete electrical circuit of Bauer indicator.

ination at the same time by pressing down a switch. Lens A and slot S_2 are carried by a horn-shaped extension of the camera.

The accurate marking of the film on top dead-centre and the beginning of the injection was obtained by utilizing the light from a mercury discharge valve which could be tripped by a minute electrical impulse. It was connected in an oscillating circuit which was interrupted after the first oscillation so as to give a well-defined flash.

Fig. 235 shows the complete circuit of the indicator camera; the circuits of the lamp and shutter mechanisms have, however, been omitted. At L_2 in this diagram is shown a smoothing choke

which works in conjunction with a condenser C_2 of 5 MF. The recharging resistance R_1 was 20,000 ohm. The two condensers C_3 and C_4 of 0·05 MF. each, charged to 6 volts through resistances of 20,000 ohm, were discharged by a contact on the fuel injector, and by a contact on the engine crankshaft, making exactly at T.D.C., respectively, through a step-up transformer, thereby providing the positive impulse for the release of the flash of light at the moment these contacts made. The rest of the diagram shows the circuit of the synchronising device S on the shaft of the mirror drum M, which is driven at one-quarter engine speed by the motor E. On the top half of Fig. 234 is indicated the manner in which the light from the stroboscopic mercury lamp L is used to print a sharp time-marking line on to the film F. The lamp L is focused by the condenser lens C_2 on the film.

Fig. 236 is a reproduction of a compression-ignition engine set of diagrams taken at 1500 r.p.m., the vertical lines showing the commencement of the fuel injection and top dead-centres. The

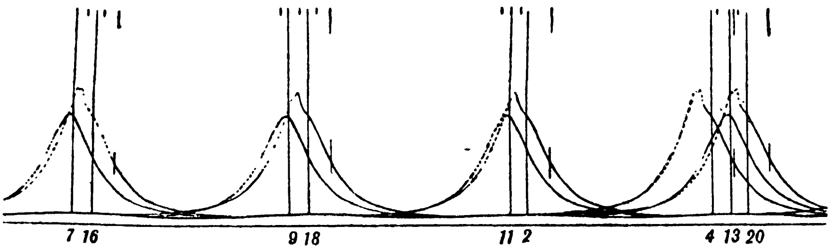


FIG. 236.—Indicator diagrams from Bauer instrument on compression-ignition engine.

numbers refer to the succession of the cycles—since several cycles can be obtained on one length of film.

Mean Pressure Indicator.—For commercial and other purposes a knowledge of the I.M.E.P. is exceedingly useful, and any instrument devised to give this quantity direct, without the necessity for measuring up indicator diagrams, would be of considerable assistance in engine test work. The effect of changes of fuel, mixture strength, ignition, speed, and other factors, is always reflected in the I.M.E.P. values.

Typical mean pressure indicators which have been made for the purpose in question include the Wimperis [1] and Geiger ones; the latter is of German origin. Both instruments were described in the second edition of this book, but as their applications to modern high speed engines have proved to be very limited they are omitted in the present edition.

[1] "On a New Method of Ascertaining the Mean Pressure in a Heat Engine," Aeronautical Research Committee's Report, No. 803, and *The Engineer*, March 2, 1923, p. 238.

Calibration of Indicators.—Apart from ensuring that the piston displacement or crank-angle movement of the indicator is exactly in phase with that of the piston or crank of the engine, the cylinder of which is under test, it is necessary to know with a fairly high degree of accuracy, the pressure-deflection scale of the piston control spring, or of the diaphragm, as the case may be. If a beam or helical spring is employed, it is usually a straightforward matter to calibrate this, *in situ*. In most indicators of the piston type, the whole indicator can be inverted (as in the case of the Hopkinson and Burstall, which have been mentioned before) and a rod inserted in the connection tube. By applying, with a stirrup or otherwise, dead weights of known value, say W pounds, the corresponding pressure per square inch can readily be ascertained by dividing this weight, W pounds, by the area, A square inches. The deflections corresponding to the loads thus applied can be recorded on the drum, or glass screen, of the indicator. Deflection readings should be taken as the loads are applied, and also as they are taken off, one by one, in order to take any backlash, hysteresis, or other similar source of error into account.

In the case of diaphragm indicators, a special gauge-testing apparatus, similar to that shown in Fig. 237, is available.

This apparatus consists of a cylinder C, in which a plunger, or piston, P, which is a very accurate sliding fit, can move up and down. The interior of the apparatus is filled with a suitable mineral oil.

The piston P has a concentric disc at its upper end, upon which circular weights, of known value, can be loaded. These weights are provided with registering projections and recesses to ensure centrality in respect to the piston. The procedure in testing a diaphragm is to connect the underneath side to the apparatus by means of a short length of copper tubing; the gauge shown in Fig. 237 is removed and the copper pipe from the indicator connected in its place.

A known weight is placed on the piston, and the screw plunger S is moved downwards in its barrel B until the weight on P just rises; the weight is given a circular motion, about its axis, to ensure the absence of "sticking"; this procedure also minimizes leakage of the oil and capillary effects.

A piece of tracing paper is placed on the ground-glass screen, and held on by a suitable set of clips, or by means of wax, and the position of the spot of light marked with a finely pointed hard pencil, for each weight that is added to the piston P, that is for each pressure. Thus the pressure scale can be determined with a high degree of accuracy.

Fig. 238 shows another form of gauge tester for both pressure and vacuum tests. Pressure tests, by dead weight, can be made up to 1000 lb. sq. in.; by means of a screw press, further pressures

up to 2000 lb. sq. in. can be measured. The vacuum gauge uses a vacuum pump and standard mercury column gauge. Shut-off values at C, D, and E enable the various tests to be made on one or both gauges or indicators.

It is necessary to take readings with both increasing and decreasing pressures, the average value for each pressure being taken. Indicators of the diaphragm type should be calibrated both before and after a set of tests. Any serious discrepancies between the two sets of reading will at once indicate either a badly clamped diaphragm, or a cracked one. The diaphragms occasionally fail by

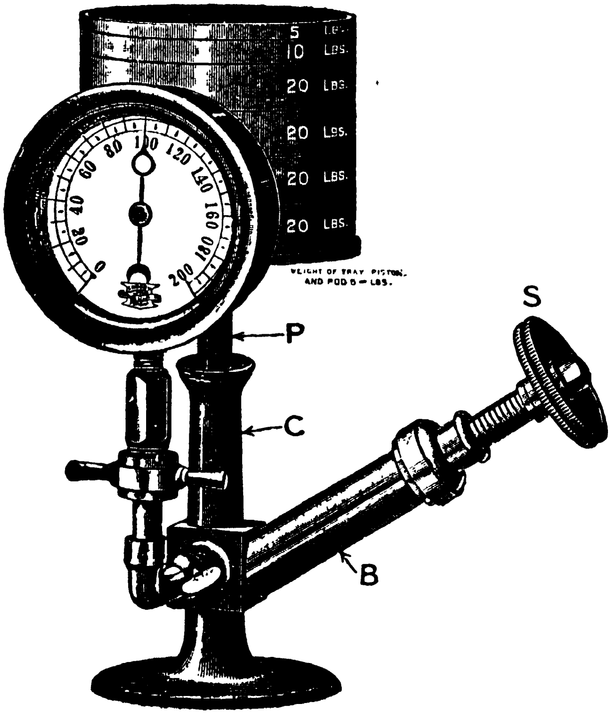


FIG. 237.—Gauge testing apparatus.

cracking, the cracks in most cases being invisible to the naked eye. They are only revealed by the oil-pressure calibration test, either by the different pressure scale (compared with the original) or by the presence of oil which, under pressure, has leaked through the crack.

The method described is a static and "cold" one. It does not simulate the conditions accurately under which these diaphragms work, although there are reasons for believing that if the temperature effect is allowed for, the hysteresis one will not introduce really serious errors.

The diaphragm can be calibrated under temperature conditions resembling the working ones, if it has a water-jacket, by passing steam or water having a similar temperature to that measured under working conditions, through the jacket, during the calibration tests. An electric resistance element heater could also be used for higher temperatures.

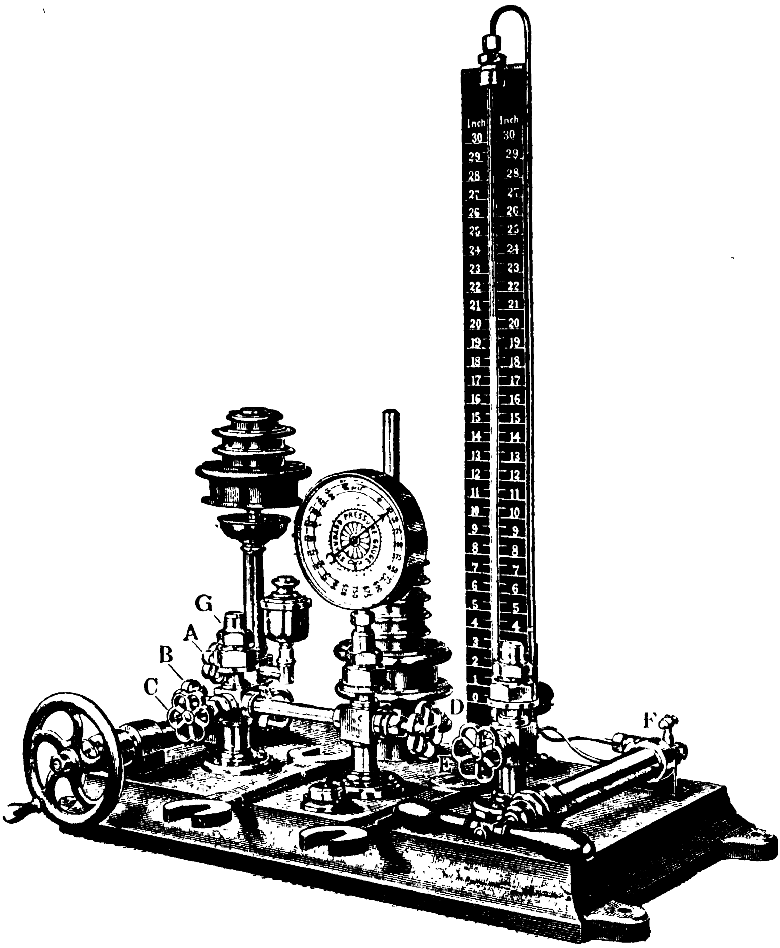


FIG. 238.—A combined pressure and vacuum tester (Budenberg).

It is possible to ascertain the mean working temperature of the diaphragm by means of a thermo-couple, and to calibrate separately the diaphragm in a testing machine in order to ascertain its pressure-deflection values at different temperatures ; alternatively a knowledge of the variation of the modulus of elasticity with temperature will enable the temperature effect to be computed from the cold calibration test.

Similarly, the hysteresis effect, due to rapid applications and removals of pressures to the diaphragm, might be ascertained in the laboratory.

Outside types of control springs, similar to those we have described, naturally keep very much cooler, and in most cases temperature corrections are unnecessary.

Temperature of Calibration.—The temperature of the connecting pipe between the engine and the indicator also has an appreciable effect upon the accuracy of the readings. If the indicator diaphragm is calibrated with the tube cold, it will give a lower deflection reading than when calibrated hot for the same pressure value. Thus, in one case [1] it was found that when the connecting tube of the Watson-Dalby indicator was calibrated at atmospheric temperature, the deflection in a certain case was 39 mm. for 85 lb. per square inch pressure. When the connecting tube was immersed in boiling water, the deflection for the same pressure rose to 45 mm. It had previously been noted that when the observed cylinder was run under power the atmospheric line obtained on momentarily cutting out the cylinder—especially with a short connecting tube—was different from that obtained after the cylinder was cut off for some time. To obviate this variation, the diaphragm was protected from the impinging hot gases by passing the pipe connections through a bath of boiling water, whether calibrating or observing, reducing the volume of the diaphragm chamber to a minimum by casting in a block of aluminium making fair contact with the water-cooled walls of the diaphragm chamber, and cooling the impinging gases on the way to the diaphragm. Three holes, $\frac{5}{64}$-inch diameter, were drilled in the casting, giving a measured capacity of 1·5 c.c. The connecting tube was 38 cm. long and 0·25 cm. in bore, having an adapter about 6·5 cm. long, giving a capacity of 3·5 c.c. The passage through the cylinder jacket was 0·35 cm. bore by 4·8 cm. long, and had a capacity of 0·5 c.c.

Indicator Cocks.—It is necessary to disconnect the indicator from the engine when the former is not required, and for this purpose a tap or cock is interposed between the two.

This cock also provides a means for connecting the indicator pressure element to the atmosphere, in order to obtain the atmospheric pressure line on the indicator diagram.

Fig. 239 illustrates a typical indicator cock, which is provided with a small hole in its shell for atmospheric pressure communication. It is nickel-plated all over, and the tap handle is made of hard fibre for heat insulation purposes. The portion Z screws into the cylinder, whilst the indicator is attached at S when the cap is removed.

[1] "Sleeve Valve Engines," E. B. Wood, *Proc. Inst. Autom. Engrs.*, April, 1923.

The disadvantages of this type are that it lengthens the passage and conducts the heat to the indicator. To obviate the latter disadvantage, Professor Burstall employs a water-cooled cock, in which both the plug and the shell are cooled. The tap in question is shown in Fig. 208, just below the piston. It is claimed that this water-cooled tap neither leaks nor wears, and that it does not stick or seize up after being in use for some time as in the case of ordinary taps.

The steel plug used is hardened, and it works in a steel shell. For many purposes the screw-in type of valve shown in Fig. 240

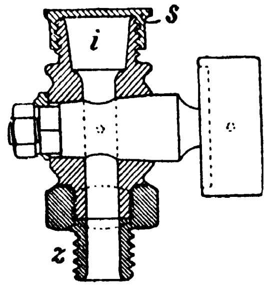


FIG. 239.

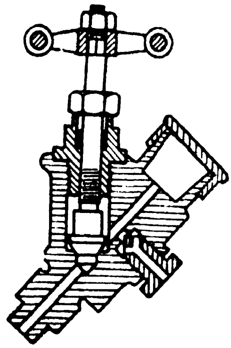


FIG. 240.

is to be preferred to the type shown in Fig. 239, more especially for internal combustion engine work, where the valve is exposed to high temperatures; a shorter length of passage can be employed in this case. The atmospheric connection is obtained by unscrewing the conical seated, internally-drilled plug shown on the right-hand side in Fig. 240.

In the (Dobbie-McInnes) "Farnboro" type indicator the balance valve unit is now arranged so as to be screwed up against the cylinder wall-bearing face, for shutting off the indicator.

Chapter VIII

CATHODE RAY INDICATORS

In recent years the speeds of petrol and compression-ignition engines have increased to such an extent that the ordinary types of indicators described previously have, with certain exceptions, proved unsatisfactory at the higher engine speeds, chiefly on account of the inertia effects associated with the moving parts of the instruments in question.

The development of public television has, however, stimulated the evolution of an entirely new type of pressure (and also movement) indicator for use at high operational speeds, namely, the cathode ray oscillograph one. The result of this attention has been the commercializing of several alternate designs of cathode ray indicators having a range of applications to problems connected with high speed engines of the types under consideration ; in this connection these indicators possess definite advantages over the other types used previously.

It should be mentioned that, unlike mechanical and optical indicators the cathode ray instrument involves electrical principles of a somewhat complicated nature, from the engineer's viewpoint, so that their installation, operation, and adjustment is a matter rather for the electrical expert if the results are to be relied upon. Further, experience with these indicators has shown that, without special electrical knowledge, the results are liable to several kinds of errors in the hands of the uninitiated.

Principle of Operation.—The cathode ray indicator consists of three units, namely : (1) the cathode ray tube, (2) the pressure element, and (3) the time base or piston position unit. The cathode ray tube, which is basically the same as that used for television purposes, resembles in outward appearance a conical glass flask on the bottom surface of which the pressure-time or pressure-piston position luminous diagram is produced. It consists of an evacuated glass tube or vessel in the narrower neck of which is a cathode plate sealed into the end and an anode plate having a central hole arranged to the left of it as shown in the diagrammatic illustration (Fig. 241). When a sufficiently high voltage, namely, from 300 to 3000 volts, is applied between the cathode and anode, an electrical discharge occurs, and since the current is carried by negative charges, and there is a steady flow of electrons from the negative to positive electrodes, these electrons are discharged at a very high velocity against the anode plate. As the latter has a central hole some of these electrons pass through this hole and travel along the tube until

they strike the end of the latter. In order to render this electron stream luminous where it strikes the tube end surface, the latter is coated with a special fluorescent material such as zinc silicate; this gives a yellowish-green spot of light where the electronic stream strikes it. For photographic purposes, however, the light in question is unsuitable, since long exposures would be necessary, so that another material, namely, calcium tungstate, is used to coat the base of the tube. This substance fluoresces with a deep blue light, having about thirty times the photographic activity of zinc sulphate; the latter is used only for visual observations.

The static beam of electrons produces only a luminous spot on the base screen, but if it is allowed to pass between a pair of plates, to which an electrical potential difference is applied in order to set up an electrostatic field, the beam will be diverted, or bent, towards the positive plate and the spot of light will, in consequence, be displaced; the amount of the displacement will be proportional to the voltage applied or potential difference between the plates.

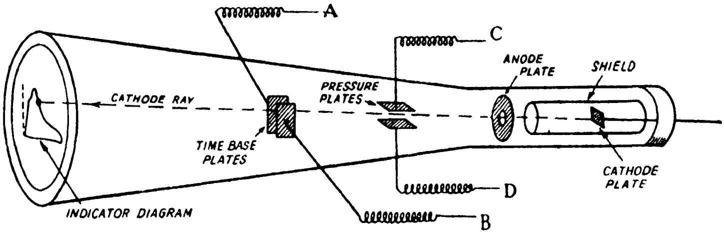


FIG. 241.—Principle of cathode ray indicator.

In the cathode ray indicator there are two pairs of plates arranged so as to be mutually at right angles, as shown in Fig. 241. One pair, known as the *Pressure Plates*, connected to the pressure element by wires C and D, is subjected to voltage variations in proportion to the gaseous pressures within the engine cylinder which are to be observed or measured. The other pair, or *Time Base Plates*, connected to its element by wires A and B, receive voltage variations proportional to the position of the engine crank, or to the time of occurrence of the observed pressures.

The spot of light is therefore displaced in two directions at right angles and, if the phase relationship between the cylinder pressure and the crank angle is correct, traces out a diagram of pressures upon a time base as indicated on the left in Fig. 241.

Notes on Cathode Ray Tubes.—It is not possible, owing to present space limitations, to enter into detailed considerations of the electrical side of cathode ray tubes, so that reference should be made to textbooks on this subject [1] and to the footnote references given in this chapter.

[1] A good elementary account is given in "Photo-Electric and Selenium Cells," by T. J. Fielding (Chapman & Hall, Ltd.), 2nd edition.

It may be mentioned, however, that a refinement which is sometimes adopted in cathode ray indicator tubes is to provide a pair of coils CC and DD (Fig. 242 [1]), mounted mutually at right angles outside the tube and opposite to the pressure and time base plates, PP and QQ, so that by passing current through these coils an electromagnetic field is superimposed upon the electrostatic field between the plates so that a means is thus provided for deflecting the electronic beam—and therefore the spot of light on the screen—in two directions at right angles. Alternatively, if the spot of light is suitably located the area of the diagram on the screen can be increased to the maximum size required.

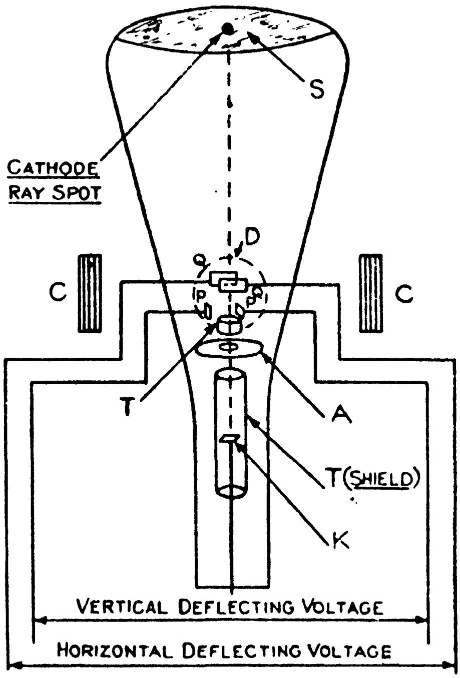


FIG. 242.—Arrangement of typical cathode ray tube.

A. Anode. C and D. Beam deflecting coils for adjustment purposes. K. Cathode. T. Metallic shield to cathode. P and Q. Pressure and time base plates. S. Fluorescent screen.

It is also usual to interconnect one of each pair of plates, as shown in Fig. 243, so that if a D.C. potential is applied to either of the two plates the deflection of the cathode ray spot on the screen is along the line AX or AY (Fig. 244). It would thus be possible to utilize one-quarter only of the area of the screen. In order to use the full area of the screen the interconnected plates are given an initial positive potential; this results in a shift of the origin A, of the co-ordinates, to the position shown at O. If, for example, the initial positive potential of the plates P_1Q_1 is, say, 80 volts, then, when a positive voltage of 80 is applied to the plates P_2Q_2, the fluorescent spot is brought back from O to A; thus, by varying the initial positive potential any desired section of a diagram can be moved to the centre of the screen.

The sensitivity of the oscillograph, which is inversely proportional to the propelling voltage of the electron stream can be varied by altering this voltage. Thus, to increase the sensitivity it is necessary

[1] " The Cathode Ray Oscillograph, A. Towle, *Proc. Inst. Autom. Engrs.*, 1938.

to provide a means of obtaining electrons at a low voltage. The cathode is therefore made in the form of a flat tungsten spiral, the heating current of which can be varied.

The fineness of the spot on the screen can be adjusted by altering both the filament current and the voltage between the anode and cathode. Excessive filament current, however, reduces the life of the tube, which is normally about 2000 hours.

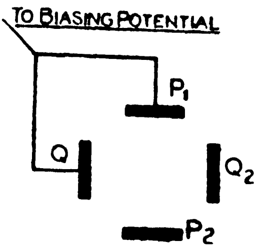


FIG. 243.

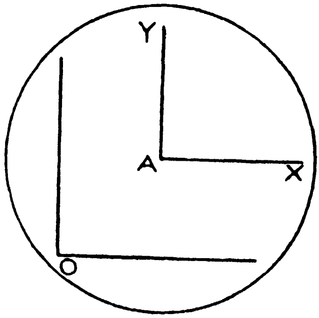


FIG. 244.

In regard to the electrical data for the usual types of cathode ray tube the following information is given :—[1]

Filament current . .	0·7 to 1·1 amperes.
Filament voltage . .	0·4 to 1·0 volts.
Anode voltage . . .	300 to 3000 volts.
Cathode to anode current .	10 to 200 microamperes.
Shield voltage . . .	0 to −200, relative to filament.
Screen deflection . .	0·75 mm. per volt across plates.

It should be mentioned that the shield shown around the cathode in Figs. 241 and 242 is for the purpose of preventing the electron stream from spreading outwards and thus to concentrate as much as possible of the stream through the hole in the anode plate ; for this purpose the shield is given a negative bias relatively to the cathode.

The cathode ray tube described is basically the same for all types of cathode ray indicator, so that no further detailed reference need be made to same.

The Pressure Element.—The purpose of the pressure element is to convert the cylinder pressure variations into corresponding voltage ones which are applied to the pressure plates of the cathode ray tube. There are several available methods for achieving this result, and it may be stated that the main differences between modern commercial indicators are essentially those of the different pressure elements employed.

[1] *Ibid.* page 286.

Broadly speaking, there are three groups of pressure elements according to whether the electrical impedance is (1) a capacity, (2) a resistance, or (3) an inductance for converting the mechanical pressure into an electrical effect. Further reference is made later to the relative merits of the individual groups. Of the methods employed four in particular appear to offer the greater advantages; these include (*a*) the moving iron or magnetophone, (*b*) the resistance method, (*c*) capacitance, and (*d*) piezo-electric systems.

(*a*) The moving iron type pressure element is contained in a plug of approximately sparking-plug dimensions screwed into the wall of the combustion chamber. The lower end of the plug has a silicon-iron diaphragm arranged very close to the end of a cylindrical type permanent magnet. The latter has an electric winding arranged concentrically, so that movements of the diaphragm due to cylinder pressure variations varies the air-gap between the diaphragm and magnet pole, thus causing changes of flux in the coil. The E.M.F. of the coil current is proportional to the change of flux and, since the reluctance of the air-gap is large compared with that of the magnet, the E.M.F. is proportional to the rate of change of cylinder pressure. The Standard Sunbury indicator employs this type of pressure element.

(*b*) The pressure element in this case consists of a cylinder of a special resistance material which possesses the property of lowering its resistance or increasing its conductivity in proportion to the increase of pressure to which it is subjected. The resistance element is indirectly in contact with a diaphragm exposed to the cylinder pressure. Thus, the variations of pressure cause similar variations of resistance which are employed to produce corresponding fluctuations of electric potential between the pressure plates of the cathode ray tube. The Cossor indicator employs this method.

(*c*) In this type of pressure element the movements of the cylinder diaphragm alter the air-gap of a condenser unit, and these variations in capacitance are arranged to produce corresponding changes of voltage across the pressure plates of the cathode ray tube. This method is used in the later model Dodds indicator.

(*d*) The piezo-electric method employs in the pressure element a number of quartz crystal plates connected in parallel, housed one above the other. When subjected to pressure variations from the cylinder diaphragm or other pressure transmitting means, the E.M.F. of the circuit containing the crystal element changes in proportion to the applied pressure. The Zeiss-Ikon indicator uses this type of pressure element.

In regard to the relative merits of these different pressure element systems there has been a good deal of controversy, from which it must be concluded that each type has its own particular advantages and also disadvantages.

The moving iron type (*a*) has a high natural frequency, namely, of the order of 40,000 cycles per second so that it is well above that of detonation pressure waves; detonating combustion conditions can, therefore, be studied satisfactorily with it. No water-cooling of the pressure element is necessary. It is easy to calibrate for the pressure scale under actual cylinder working conditions. The effect of temperature on the pressure unit is practically negligible. The indicator, with suitable changes of pressure element units, can be used for observing the pressure variations in compression-ignition fuel systems and also the movements of the fuel valve itself. Inlet pipe and weak spring indicator diagrams can readily be obtained. Engine vibrations due to out-of-balance effects and combustion shocks can also be studied in detail. It is further possible to obtain rate-of-pressure change diagrams as well as the ordinary direct pressure ones on a time-base.

The resistance element type (*b*) gives satisfactory results, but as the diaphragm is subject to fatigue and corrosion effects it is necessary to water-cool the pressure element. The calibration of the pressure scale is not a difficult operation, but it has been found that when used over appreciable periods the calibration values tend to change. It is apt to give unreliable results under detonation or similar shock conditions and cannot conveniently be adapted to some of the other types of measurement, such as engine movements and vibrations, which are possible with type (*a*). For these reasons this type of pressure unit has been to a large extent replaced by types (*a*) and (*c*).

The condenser type pressure element (*c*) gives satisfactory diagrams and does not require water-cooling. It can be used for both direct pressure and rate-of-pressure change diagrams.

The piezo-electric pressure element (*d*) is particularly suitable for high frequency pressure variations. It is also covenient for light spring diagrams, such as crankcase and inlet pipe pressure diagrams. On the other hand, its uses appear to be limited to pressure-time observations, for it is not suitable for rate-of-pressure diagrams or therefore for detonating conditions in internal combustion engines. For purely pressure-time records the piezo-electric indicator is probably the best of any.

Owing to its advantages for high frequency effect measurements this type is well suited to the study of detonation, impact, explosions, shock, and similar rapid transients. The crystal should be connected by a cable of low capacity and high insulation resistance to the cathode ray tube.

For medium-frequency effects the moving iron, or magnetophone, type has certain advantages, whilst for low-frequency effects and for steady and varying pressures the moving iron type energized by high-frequency current of, say, 20,000 cycles per

second, is recommended. Medium-frequency effects up to 2,000 cycles per second are also recorded by this system.

In general the whole field of high speed internal combustion conditions appears to be covered by the piezo-electric, capacitance and moving iron pressure element types, with steady excitation and with high-frequency alternating current excitation as alternatives.

Typical Cathode Ray Indicators.—Since the possibilities and advantages of this type of indicator have been recognized a good deal of experimental work has been carried out with alternative designs, more particularly in regard to pressure units and electrical circuits, with the result that several commercial indicators have been developed in this country, in the United States and Germany. As it is not possible owing to space considerations to describe all of the various types now available, a well-known instrument will be described and its applications pointed out. The special features of other designs will also be referred to.

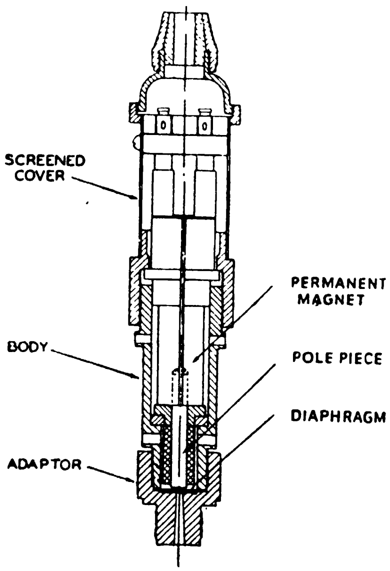


FIG. 245.—Pressure element of Standard Sunbury indicator.

The Standard Sunbury Indicator.—This indicator is one of the most successful types yet developed and it has a relatively wide range of applications.

The pressure element, illustrated in Fig. 245, is based upon the magnetophone or moving iron principle described previously, and consists of a stainless steel plug with a 14 mm. or 18 mm. sparking-plug thread for screwing into a sparking-plug hole in the combustion chamber. It contains a silicon-iron diaphragm about 0·030 inch thick which is held firmly on a seating, the underside communicating with the combustion chamber by means of a small bore hole ; this arrangement counteracts the action of the high temperature gases. A magnetic pick-up screws into the shell of the plug and is so positioned that the pole of the permanent magnet is a few-thousandths of an inch from the upper face of the diaphragm ; it is located and secured in position by means of a lock-nut. The two ends of the magnet encircling coil are connected by means of suitable leads to two terminals near the top of the plug. The diaphragm has a natural frequency of about 40,000 cycles per second, and there is a slight movement, namely, from 0·0003 to 0·0005 inch under normal petrol

engine pressure conditions; it has therefore a very low inertia effect.

It should be mentioned that it is difficult to make such a pressure unit with a length of small bore less than about 20 mm., so that there is a certain time-lag between the moment of occurrence of pressure in the cylinder and its indication by the pick-up unit; this time-lag is of the order of 35 micro-seconds. Apparently there is no diagram distortion due to this effect, but it has been shown that too large an air space between the small bore connection and diaphragm magnifies the effect of detonation on the rate of change of pressure diagram. This drawback is overcome, however, by the use of a plug of the design shown in Fig. 246. This has five $\frac{1}{16}$-inch holes drilled half-way towards the diaphragm from the cylinder. These holes open out into an air cell of $\frac{3}{8}$-inch diameter by 0·015 in depth, and are followed by fifteen $\frac{1}{32}$-inch holes opening into an air cell 0·005 inch

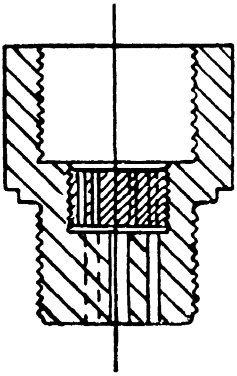


FIG. 246.—Design of plug to reduce time-lag effects.

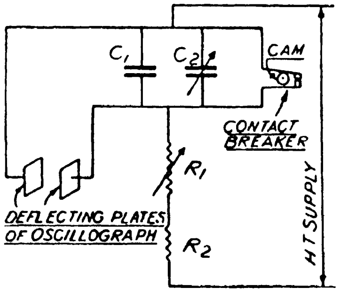


FIG. 247.—Time-base circuit.

deep under the diaphragm. This arrangement is such that there is no direct connection between the cylinder and the diaphragm.

The pressure element described is for cylinder pressure diagrams, but other patterns are made for compression-ignition fuel line and spray valve units.

The Time-base Unit.—This consists of a special type of contact-breaker having its contacts mounted on an adjustable ring which can be rotated at will through a complete revolution. These contacts are operated by a cam driven from the engine crankshaft. The electrical circuit employed is such that rotation of the end-plate of the contact-breaker unit controls the starting-point of the diagram on the cathode ray tube screen. The contact-breaker is connected in series with a small resistance, and the two are connected across a variable condenser consisting of a fixed condenser C_1 (Fig. 247) and a variable one C_2. The condenser unit is connected in

series with a variable high resistance R_1, a fixed resistance R_2, and a source of high voltage supply. The contact-breaker remains closed for about 15° of crank angle after which it opens and the condenser is then charged through the variable resistance, and the increasing voltage—which is very nearly linear in respect to time—is applied to the time-base or deflecting plates of the cathode ray tube (marked "oscillograph" in Fig. 247). The spot of light therefore moves across the screen giving a horizontal sweep at a rate which is determined by the value of capacity and resistance in the circuit. When the contacts close the spot is brought back to its equilibrium position and the process is repeated. With the values of capacity and resistance provided, the whole of the cycle down to as little as 30° of crank angle at 3000 r.p.m. can be made to fill the screen. The later designs of contact-breaker are very light in construction and can be used for engine speeds up to 10,000 r.p.m.

There is also a *time-sweep* device embodied with the contact-breaker unit for the purpose of producing degree marks on the diagram to calibrate it for crank angle and also to provide the contacts which "trigger" the time-base of the cathode ray tube (as shown by the sinuous line marked with degrees in Fig. 252).

The degree marks are produced by a steel disc, the rim of which is slotted at 2° intervals, with deeper slots at each fifth tooth and deeply cut slots at 90° intervals. A small permanent magnet provided with a winding is fixed with its pole close to the rim, and the alternating voltage generated by the rotation of the disc, which is mounted on a spindle and driven through a coupling from the engine crankshaft, is applied to the input of the amplifier. The corresponding figure obtained on the cathode ray tube consists of a serrated line across the horizontal diameter of the screen. Every two, ten and ninety degrees are thereby marked, and as one of the deep 90° slots in the disc can be quite simply aligned at top dead-centre, the crank angle positions along the indicator diagram can be directly determined.

The Complete Indicator.—The complete Standard Sunbury indicator (Fig. 248) consists of (1) the pressure element or pick-up unit, (2) the time-base and calibration unit, (3) the amplifier, (4) the cathode ray tube unit, and (5) the low-tension accumulator. The diagrammatic layout shown in Fig. 248 refers to a single cylinder compression-ignition engine with pick-up units for indicating cylinder pressures, fuel injection pressures and injection valve movements.

The Amplifier.—This unit consists of a three-stage resistance-capacity coupled amplifier, which uses triodes in all stages and is specially designed to amplify uniformly over a very wide range of frequencies, from less than one up to several thousand cycles per second. The input is controlled by four switches which enable

any one of the three engine pick-ups or the degree marker to be selected as required. Three additional switches are provided, the first of which allows the use of either two or three stages of amplification, the second controls the use of the integrating circuit to convert rate-of-change of pressure to direct-pressure diagrams, and the last enables the meter to read either the filament current of the cathode ray tube or the low-tension accumulator voltage. Control of the amplification is arranged by an adjustable sensitivity control having ten equal steps each corresponding to 10 per cent. reduction in overall magnification, so that diagrams reduced in a known proportion can be obtained. A compensation control is provided to eliminate the zero line error, the normal position being marked on the scale. Coarse and fine controls enable the time

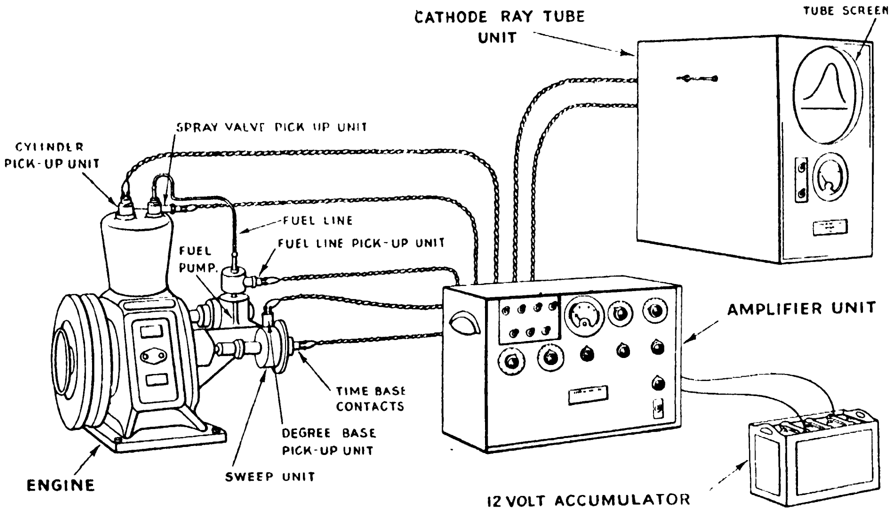


FIG. 248.—The Standard Sunbury indicator.

scale to be expanded or contracted, and the usual filament rheostat and focus control for the tube, and spot bias control for both vertical and horizontal directions are included. All the components are assembled on a metal chassis which is held in a metal containing case, and all connections are made at the rear of the case by means of plugs and sockets which are clearly designated. The only external power supply required is obtained from a 12-volt accumulator of about 50-ampere-hour capacity, the anode current for the third valve being provided by a small H.T. motor generator mounted inside the amplifier and driven from the 12-volt accumulator. The H.T. supply for the first two valves is derived from a set of dry batteries contained in the tube unit. The drain on these batteries is extremely small, and consequently they rarely require replacement.

With the usual 7-inch diameter cathode ray tube a voltage of about 100 is required to give the full-scale deflection on the screen. Since the voltage given by the pressure element is about 0·5 volt an amplification of 200 is needed for the rate of change of pressure diagram.

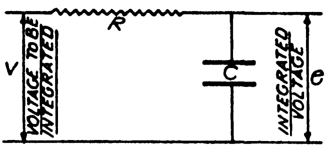


FIG. 249.

The Integrating Circuit.—It should here be explained that the electrical output obtained from the pressure element is proportional to the rate of change of pressure on, or velocity of, the diaphragm. The diagrams thus obtained are therefore velocity or "rate of change" of pressure ones. In order to

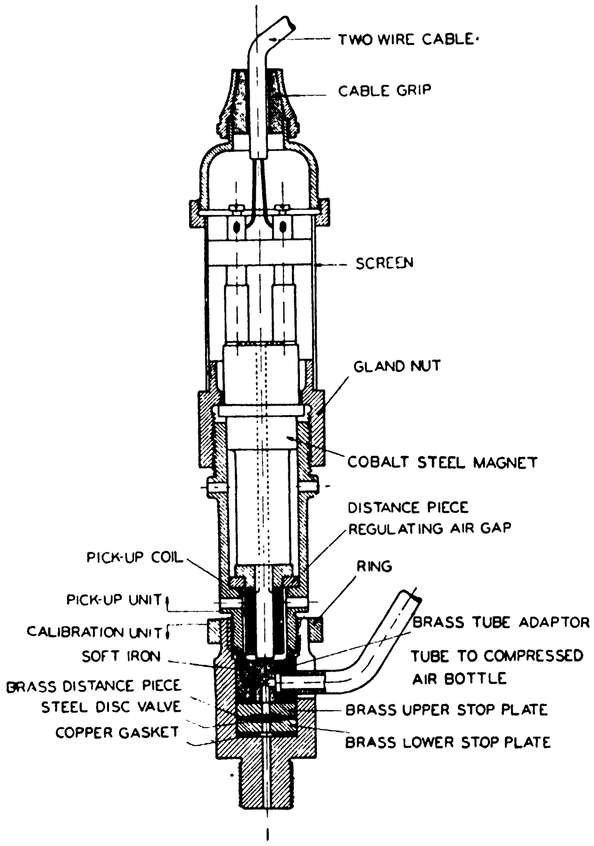


FIG. 250.—Method of calibrating pressure unit.

convert these into pressure-crank-angle (or time-base) diagrams it is necessary to integrate the velocity diagram. This integration is performed electrically by means of a special integrating circuit

(Fig. 249), included in the second stage of the amplifier, which can be thrown into or out of use by a switch. The integrator circuit consists of a resistance R and condenser C of such values that the output voltage is a measure of the time integral of the applied voltage. The integration by this circuit alone is not quite perfect and a feed back control is provided, which exactly compensates for the inherent inaccuracies. By these two simple devices, i.e. the integrator and feed back circuits, either velocity or pressure diagrams can be obtained at will, thus very greatly extending the value and scope of the indicator.

Calibration of the Pressure Element.—There are several methods of calibrating the indicator, the one selected depending upon the degree of accuracy required and the particular pick-up unit employed.

For cylinder pressure measurements it is usually sufficient to apply a maximum pressure gauge to the engine in place of the indicating unit, and to take a reading from this under given engine conditions. Observation of the maximum height of the indicator diagram under the same engine conditions gives the required calibration, and, assuming linearity, an error of not more than 2 per cent. of full pressure will be made at any part of the diagram.

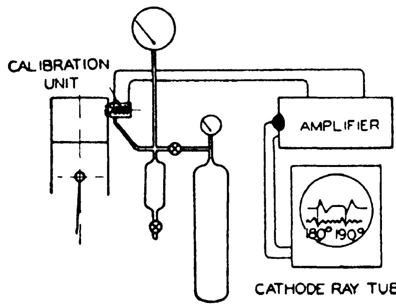


FIG. 251.—Pressure unit calibration method.

Alternatively, a somewhat similar method, based upon the use of air supply of variable pressure on the lines of the "Farnboro" indicator system can be employed.

Fig. 250 illustrates the type of calibration unit used on the Standard Sunbury indicator and shows a steel disc valve of about $\frac{3}{16}$-inch diameter and 0·020-inch thickness which can move up and down between two seats located 0·002 inch apart.

Air pressure is applied to the upper side of the disc whilst the lower side is exposed to cylinder pressure (Fig. 251). At the moment the cylinder pressure exceeds the balancing pressure, the disc lifts sharply and remains on the upper seat as long as the cylinder pressure exceeds the balancing pressure. The moment the disc valve lifts, it is registered as follows : Instead of using an electric contact, the disc, acting as an armature, cuts magnetic lines and generates voltage in a coil wound around the magnet. This offers two distinct advantages. The signal is recorded, not when the movement (0·002 in.) is completed, but when it commences, thus avoiding any delay. The method in question eliminates the use of electric

contacts which are liable to arcing and pitting effects. When connected to the amplifier unit and cathode ray tube the diagram obtained consists of a straight base with sharp peaks at each balancing point. By means of the degree scale the angular positions of these peaks can be ascertained and the readings of the pressure gauge (Fig. 251) give the corresponding pressure. By gradually increasing the air pressure and making observations of the degree positions the whole of the indicator diagrams can be calibrated. Normally, there are two points on the diagram where the pressures are the same, namely, one on each side of the diagram. As the air pressure is increased these points move up the compression and expansion lines towards each other and eventually meet at the maximum pressure value.

As the disc valve weighs only about 0·03 grain the inertia effects are so small as to be almost negligible.

Fuel system pressure calibrations can be made by testing the fuel valve for static lift pressure, and then using the valve with the engine running slowly with the least possible fuel. Using the fuel-line pick-up unit close to the valve, the diagram shows clearly the opening point of the valve and direct measurement gives the fuel pressure scale.

In the case of spray valve lift diagrams, it is generally sufficient to run the engine at a load and speed at which the valve is seen definitely to reach a stop, and knowing the maximum possible lift, to assume linearity.

Temperature effects on the indicating units can be neglected in most cases as the diaphragms operate at temperatures between 100° and 180° C., and the expansion between the casing and the magnet which alters the air-gap is negligible. Cases may arise where calibrations are required to meet abnormal conditions, and these must necessarily be dealt with specially.

Typical Applications of Standard Sunbury Indicator.—The indicator can be used, with suitable pick-up units and accessories for the following purposes: (1) Cylinder pressure-time diagrams, (2) Cylinder rate-of-pressure diagrams, (3) Weak spring diagrams of the suction and exhaust pressures, (4) Weak spring diagrams of inlet pipe pressures, (5) Exhaust pipe pressure diagrams, (6) Compression-ignition fuel injection pressures and rates-of-pressure diagrams, (7) Fuel injector valve lift diagrams, (8) Engine vibration movements on a time-base, (9) Cylinder strain diagrams.

Fig. 252 illustrates a typical *compression-ignition engine diagram* showing the actual cylinder pressures, the rate of pressure changes and the time-base (crank-angle degrees) calibration record. The diagram has been reproduced approximately one-half the actual size obtained on the cathode ray screen. The rate of pressure diagram shows rises of pressure downwards, and it will be observed

that the point of change of curvature is much more emphasized than in the normal pressure diagram above. The crank-angle sinuous line shows the smaller 2° peaks, the larger 10° ones, and the still larger 90° peak on the top dead-centre line.

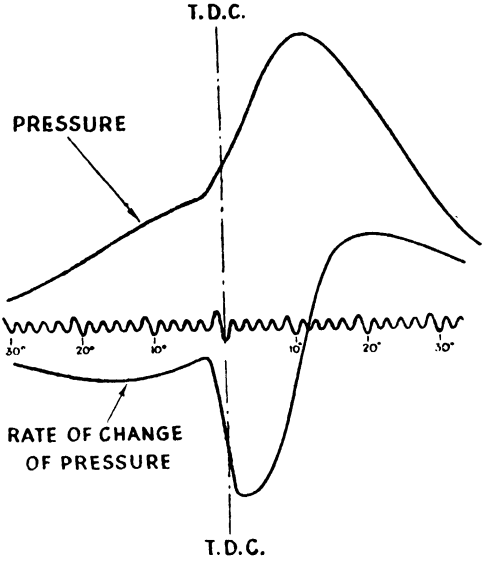


FIG. 252.—Typical compression-ignition engine diagrams (½ scale).

Fig. 253 shows two *cylinder pressure diagrams* obtained *from a petrol engine* running smoothly and under detonation conditions; it should be mentioned that the point of ignition was picked up electrostatically, without the use of special connections. The detonation portion of the curve can be analysed in much greater detail by means of the rate-of-pressure diagram used on an extended time-base arranged to cover the corresponding crank angle over which detonation occurs, namely, about 10° in the example illustrated in Fig. 253.

Fig. 254 (above) [1] shows a pressure diagram from a petrol engine running under detonation conditions. The lower diagram shows the first part of the rate of pressure rise corresponding to the upper diagram, by the full lines. In these diagrams it will be observed

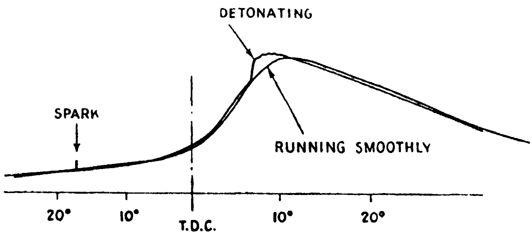


FIG. 253.—Petrol engine diagrams.

that detonation does not start until slightly after top centre, although the knock was heavy, and the rapid pressure rise takes place in only two to three degrees of crank angle being followed by pressure changes at a relatively high frequency and also by a small further

[1] "The Standard Sunbury Engine Indicator," E. S. L. Beale and R. Stansfield, *Engineering*, Dec. 20, 1935.

rise of pressure. The smooth running diagram obtained after changing to a fuel of much higher octane number is shown by a dotted line.

The first part of the corresponding velocity diagram (or rate of pressure rise) is shown below in Fig. 254 in full lines, positive rates of change of pressure being below the base line. Note the waves of very small amplitude just before the detonation starts. These have a frequency which agrees in order with that expected from the short length of indicator passage and they afford a means for correcting for the phase lag due to the passage, a correction of one-quarter of the wave-length being necessary.

Detonation produces a vibration of high amplitude at the frequency of a sound wave backwards and forwards in the engine cylinder. The first eight periods of this wave are shown followed by dotted lines giving the average envelope of the wave amplitudes. The chain dotted diagram was taken from a petrol of different

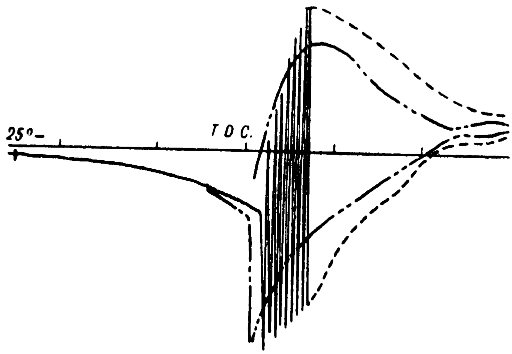


FIG. 254.—Pressure diagrams from petrol engine under detonating conditions.

chemical type. Note that the burning becomes slightly more rapid before detonation and that detonation starts earlier in the cycle, also that the amplitude of the vibrations is much less (shown by the chain dotted envelope). The two fuels indicated had precisely the same octane number measured by a bouncing-pin indicator, the pin being used in the same position in the cylinder as the electrical indicator.

A typical example of a *pressure diagram* taken from the *fuel feed line* of a compression-ignition engine is given in Fig. 255. The pressure element or pick-up was arranged close to the spray valve. The serrated horizontal part of the record at a pressure of about 80 atmospheres occurs at the beginning of opening of the spray valve, and the closing of the valve comes just beyond the last well-defined peak on the right. The surge of pressure about 70° after top dead-centre is due to a shock wave set up by the rapid flow of fuel through the pump suction port, when this is opened on the downstroke of the plunger.

Fuel pressure diagrams taken at the pump end show higher pressures by amounts depending on the pipe length and size and the viscosity of the fuel used, and the wave forms are displaced to the left by the speed of sound in the length of the pipe between the respective positions of the indicating units.

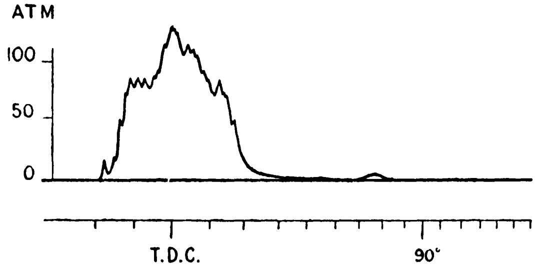


FIG. 255.—Fuel feed line pressure diagram.

Typical *spray valve lift diagrams* are valuable in connection with comparisons of the ignition qualities of various fuels, the delay angle between the moment of opening of the fuel injection spray valve and the beginning of combustion being measured. For this purpose it is necessary to measure the lift of the spray valve at different crank-angle positions. Fig. 256 shows a spray valve lift and velocity diagram taken at an engine speed of about 600 r.p.m. and at low load.

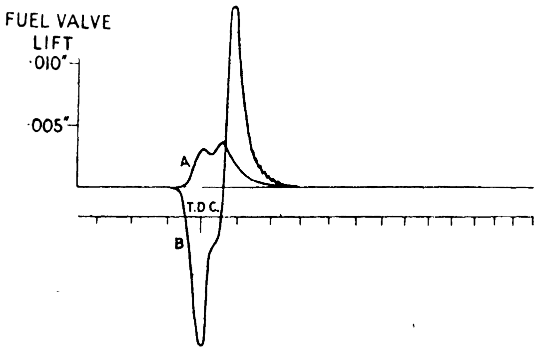


FIG. 256.—Spray valve lift and velocity diagrams.

Observation of spray valve behaviour often shows that there are several critical combinations of load and speed at which the point of opening may jump erratically from cycle to cycle. Sometimes the valve may lift, drop, and then lift a second time. This has been the case in the lift diagram (A) illustrated, but in the velocity diagram (B) this effect did not occur while the record was being taken. The clear indication of the opening point in the velocity diagram will be noticed, and also the vibrations of the whole valve

assembly, where the rapidly closing needle strikes and squeezes out the oil film on the seating.

Weak spring or *low pressure diagrams* can be taken either by reducing the air-gap between the magnet pole-piece of the pick-up and a standard thickness diaphragm, so that the deflection becomes so large that a weak spring diagram is obtained and the high pressure portions are completely off the screen, or, alternatively, a thinner diaphragm can be used and the gap adjusted so that at the higher cylinder pressures the gap is closed and the pole-piece acts as a limit stop.

Fig. 257 shows a typical *weak spring diagram* obtained from a compression-ignition engine running at 1000 r.p.m. It gives a

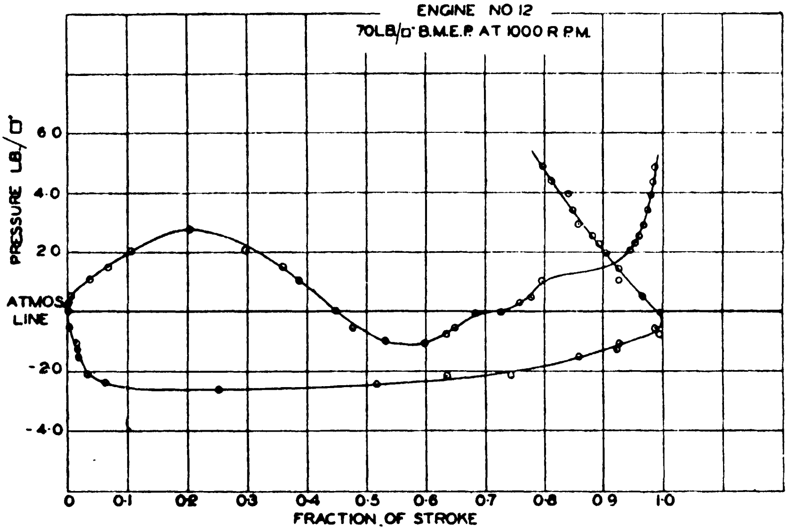


FIG. 257.—Suction and exhaust pressure weak spring diagram.

relatively large scale of pressure ordinates which enable the pressure variations to be studied in detail and accurate measurements obtained.

An example of *inlet pressure diagrams* obtained in the inlet pipe of a four-cycle engine near the inlet port is given in Fig. 258. In this example the pipe was 3 feet long and the engine speed 900 r.p.m. The opening to suction involves acceleration of the air column in the suction pipe and the consequent negative pressure is shown by the large downward wave of the diagram. The pressure then rises almost to atmospheric, falls a little as the piston reaches maximum velocity, and then rises above atmospheric pressure towards the end of the suction stroke as the moving air column in the pipe is arrested. When the suction valve closes, the air in the pipe is set in vibration at the natural frequency due to the pipe

length, the amplitude falling slowly with frictional damping until the next suction stroke starts.

These inlet pressure diagrams have proved useful in connection with the design of inlet manifolds of two-cycle engines which utilize

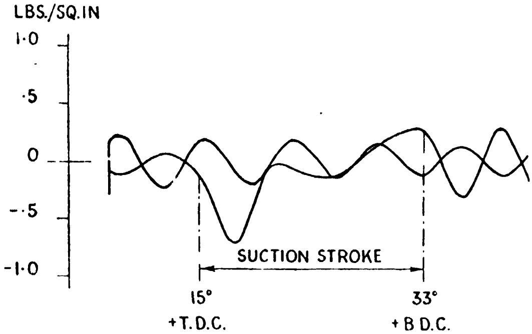


FIG. 258.—Inlet pipe pressure diagrams.

the resonance effect of the inlet pressure variations to create a suction at the moment of opening the exhaust valve in order to obtain a better volumetric or charge efficiency.

The *critical speeds* and the angular deflections proportional to crankshaft stresses can also be measured; the method used consists in driving an additional degree marker disc through a flexible drive from the shaft under examination, and connecting the output of its pick-up in series with the pick-up on a similar degree marker fixed rigidly to the shaft. Instantaneous departures from the mean angular velocity immediately become apparent, and measurements of the maximum deviation give the critical speed. Corresponding stresses are obtained by calculation.

Any movement of a part of an engine or machine can usually be arranged to cause a similar movement of a magnetic member in the field of an electro-magnetic pressure-element unit which can then be integrated in the cathode ray indicator to give a diagram showing the displacement on a time-base of the part in question.

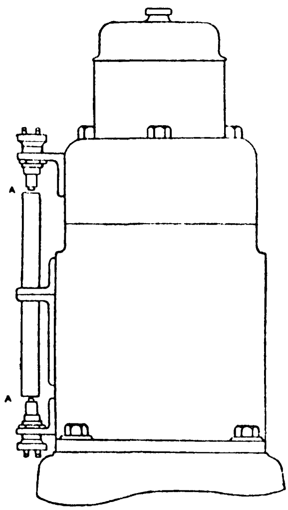


FIG. 259.—Engine movement measurements.

This method, which is similar to that previously referred to for fuel spray valves can be applied to internal combustion engine cylinders by mounting the pressure element near to the vibrating

part in such a manner that no vibration is transmitted directly to it. An arrangement for measuring strains and vibrations due to combustion pressures and shocks is shown in Fig. 259.[1] In this application any movement of the engine as a whole will not affect the air-gaps A, and only the internal strains in the cylinder will give rise to a variation in the gap so that movements due to detonation knocks can readily be studied.

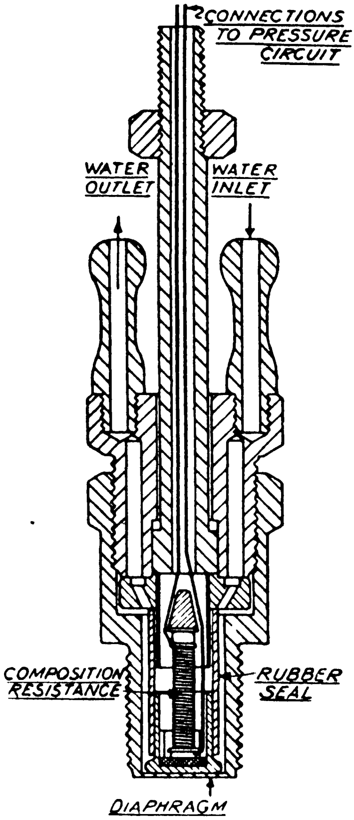


FIG. 260.—Pressure unit of Cossor indicator.

Engine vibrations under non-detonation conditions due to lack of proper engine balance can be studied by mounting the pick-up unit near to the cylinder head but in such a manner—as previously mentioned—that no vibration is transmitted to it. The air-gap between the fixed unit and vibrating cylinder head can then readily be observed with the cathode ray indicator.

The Cossor Indicator.—This indicator, as mentioned previously, employs a special resistance type of pressure element, the conductivity of the resistance material increasing as the pressure rises. The general lay-out is shown in Fig. 260 from which it is seen that the unit is water-cooled.

The complete indicator is shown in Figs. 261 and 262. A simple time-base device is embodied which operates in conjunction with a contact-breaker worked off any convenient member of the engine such as a cam or the valve rocker arm.

The indicator has also a piston-displacement arrangement for giving indicator diagrams on *a piston stroke base*. A somewhat similar device is employed on some of the other proprietary makes of cathode ray indicator. The complete piston-displacement apparatus consists of two separate units, namely, a one-valve amplifier and electrical controls, and a photo-electric cell member. The principle of this device is as follows :—

A cylindrical cam having a special profile is attached to the engine shaft or cam shaft. It revolves freely inside the unit. The

[1] *Ibid.* page 297.

FIG. 261 The Cossor indicator.

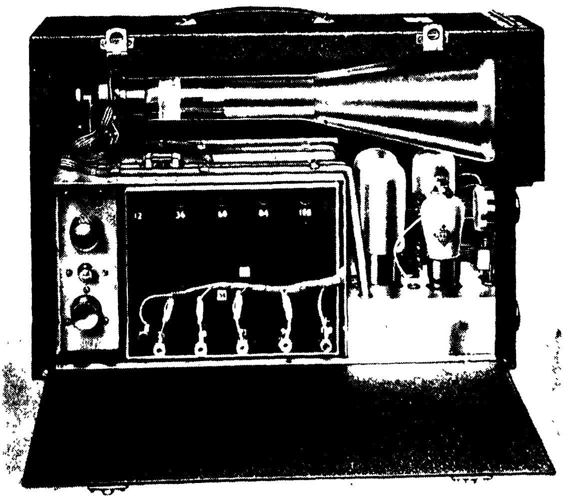


FIG. 262 Interior view of the Cossor indicator.

[To face page 302.

FIG. 263. Indicator diagram taken with ignition fully retarded, 1200 r.p.m.

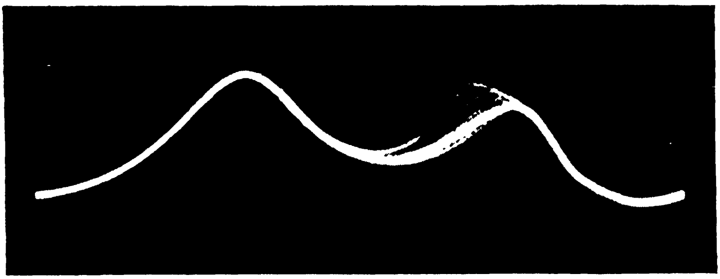


FIG. 264. Indicator diagram taken on a time base.

[*To face page* 303.

profile of the cam is so cut that it controls the area of light passing from a small lamp to a photo-cell in such a manner that the area of light open for that cam position causes the cathode ray spot to indicate the piston position at that instant.

The one device can be used to duplicate on the screen the motions for all the pistons of a multi-cylinder engine, in addition to permitting out-of-phase motions. For engines having different ratios of connecting rod to crank, it is necessary to have different cams, but since the cams are easily cut this is no drawback. With a suitably shaped cam it is also possible to obtain a crank-angle base, and of course, one cam will then be suitable for any engine. One valuable feature of the crank-angle cam is that it can be arranged to make the spot move across the screen slowly for, say, 340° rotation of the engine shaft, and then to fly back to the start quickly, so that the return traverse of the spot represents the remaining 20° rotation of the engine shaft. In practice the 360° can be divided into any two fractions—say, 350° + 10° or 300° + 60. The return sweep of the spot can be made as low as 7° rotation of the engine shaft even with the standard equipment.

The box contains a one-valve amplifier and two controls ; one to alter the length of the diagram and the other to position the diagram on the screen. In operation the box must be placed near to the engine.

The pressure unit has a 14 mm. thread and screws into a sparking-plug threaded hole. The pressure unit is calibrated by means of a special grease-gun screw-in plunger and pressure gauge, the pressure unit screwing into the adaptor body of the latter, and electrical connections are made to the cathode ray unit. The " gain " control is given a definite setting and the cathode ray spot is moved to the bottom of the screen. The pressure on the pressure element is increased by means of the grease-gun screw by equal increments, and each new position of the spot is marked on the screen itself, photographed or traced on paper held against the screen by means of a special fitting supplied for the purpose.

The advantages of this device are numerous. Its attachment to the engine is simple, and there are no rubbing or wearing parts. It is unaffected by vibration and the length of the diagram is independent of engine speed ; further, there is no need for manual operation of the controls to keep the diagram on the screen.

Fig. 263 is a reproduction of a typical indicator diagram taken from an engine running at 1200 r.p.m. with the ignition fully retarded ; the corresponding diagram on a time-base is shown in Fig. 264. The diagrams are reduced in reproduction, the actual widths being about 3¾ inches.

The Dodds Indicator.—This indicator employs a resistance type of pressure unit (Fig. 265 [1]) consisting of a body A machined

[1] *Automotive Industries.*

from the solid in stainless steel. Screwed into the stainless body is a boss B, which carries a central electrode C, suitably insulated by means of silica tubes D. The lower enlarged end of this electrode is ground flat and contacts with the upper surface of a stack of thin carbon discs E, about $\frac{3}{16}$ inch in diameter and $\frac{3}{16}$ inch high. This assembly is shrouded by means of the rubber tube F, allowing the complete element to be water-cooled without interfering with the electrical circuit. Initial adjustments are made by screwing in the member B, which is locked in position by the locknut G. As a result of water-cooling, temperature "creep" of the resistance element is eliminated, and, at the same time, the mechanical properties of the stainless steel are preserved.

The pressure element is connected in circuit with deflecting coils on the neck of the cathode-ray tube. There is no need for amplification, and the height of the diagram obtained is a function of the battery voltage. The timing mechanism, which is driven from the crankshaft, is similar in design to an ignition breaker, but involves a separate set of contacts on the upper side of the breaker arm. One or the other pair of contacts is selected by moving the complete stationary contact around the axis of the contact-breaker arm, thereby ensuring good contact operation at all settings. When the conventional pair of contacts is in use, the diagram obtained represents 350° of a complete rotation, the remaining 10° corresponding to the period of "make." When the upper pair is in use, only a small portion, 20° for instance, is selected, the contacts remaining closed during the other 340°. The breaker mechanism is connected to a timing-base unit consisting of a condenser which is charged at any desired rate by means of a variable resistance, and short-circuited once each rotation by the breaker. Two neon tubes are connected in series across the condenser to prevent the attainment of dangerous voltage between the plates of the condenser.

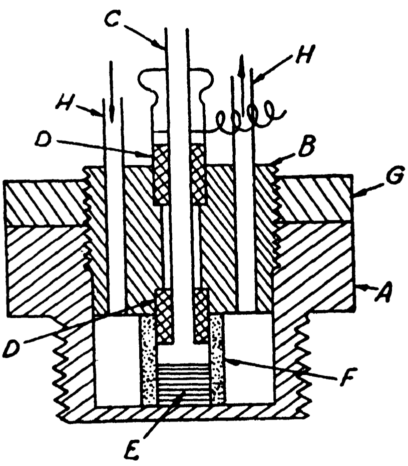


FIG. 265.—Dodds indicator pressure unit.

In a later design of the Dodds indicator the stainless steel sparking-plug body is blanked off at the end by means of a thin diaphragm of about 0·010 to 0·050 in thickness, and between the diaphragm and the central electrode there is a thin sheet of mica. The pressure unit operates on the variable capacity principle whereby any change

of pressure on the diaphragm varies the thickness of the mica and thus alters the value of the capacity between the diaphragm and central electrode; this change is employed through the agency of a sensitive electrical circuit to produce a corresponding voltage change across the pressure plates of the cathode ray tube. By means of a two-way switch either a pressure or rate of pressure change diagram on a time-base can be obtained. In one position of the switch a circuit which is brought into action, has a high

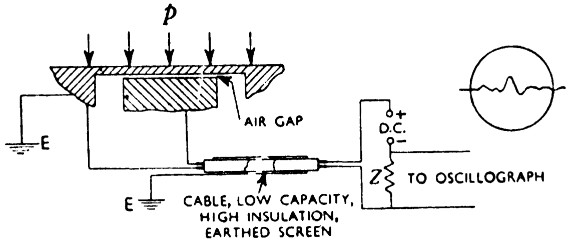


FIG. 266.—Principle of condenser type pressure element.

time constant, and is arranged so that the potentials applied to the amplifier are proportional to the capacity of the pressure element. In the second position of the switch another circuit is employed having a low time constant such that the potentials applied to the input grid are proportional to the velocity of the diaphragm, that is, to the rate of change of pressure on the diaphragm; no water-cooling is necessary for this type of pressure element, but an external copper cooling coil is employed.

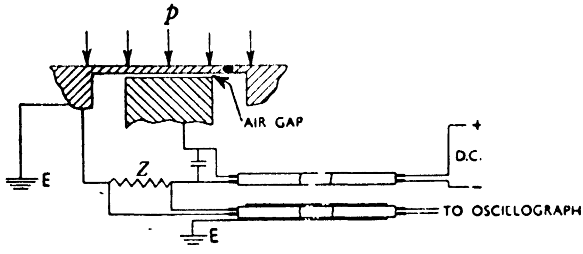


FIG. 267.—Condenser type pressure element.

It should be mentioned that the latest indicator is known as the Cossor-Dodds, this being the Dodds indicator as manufactured by Messrs. Cossor Ltd., in place of the Cossor indicator described.

Condenser Type Pressure Elements.—The general principle of the condenser type of pressure element is illustrated in Figs. 266 and 267.[1] The diaphragm above is exposed on the combustion chamber side to a pressure p which varies the air-gap shown and

[1] "Basic Principles of Cathode Ray Oscillograph Engine Indicator," F. D. Smith and E. H. Lakey, *Proc. Inst. Mech. Engrs.*, Dec. 1939.

20

therefore the capacity between the upper and lower shaded elements, one of which is earthed at E. If a steady electromotive force is applied to this condenser unit and to another electrical impedance in series with it the condenser voltage changes are proportional to the cylinder pressure changes. These voltages of the condenser are amplified and applied to the pressure plates of the cathode ray tube. The capacity of the cable, which must be kept as low as possible, cannot usually be kept below that of the pressure element so that a circuit of the type shown in Fig. 267 is employed. In this case an impedance Z, which is small in comparison with that of the air-gap, is connected directly in series before transmission through the cable.

The Piezo-electric Indicator.—The principle of this type is illustrated in Fig. 269 from which it will be seen that the piezo-electric crystal—which is usually made of quartz—is subjected to the cylinder pressure changes p from the diaphragm shown above. The corresponding variations cause changes in electric

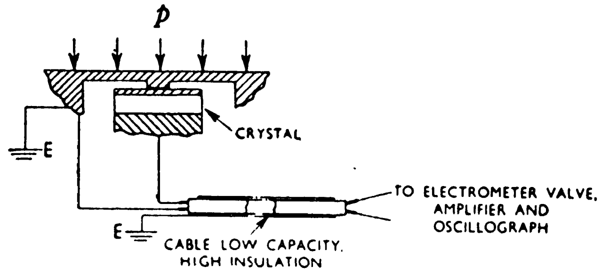


FIG. 268.—Piezo-electric pressure unit.

potential across the crystal, and these are amplified and applied to the pressure plates of the cathode ray tube. The device has a high electrical impedance, so that a short low-capacity cable must be used, as with the condenser type pressure element. This type is considered the best for high-frequency records.

Typical indicators employing quartz crystals are the Zeiss-Ikon (Germany), the R.C.A. (America), the Kluge and Linckh (Germany), and the Watson and Keys (Canada).

The Zeiss-Ikon indicator employs a number of quartz discs cut from rock crystals, and the complete unit resembles a sparking plug. In order to enable suction-stroke and inlet manifold diagrams to be taken, the quartz pile is normally subjected to a slight pressure. The time-base unit, which is shown below on the right in Fig. 269, gives a piston-stroke base diagram of pressures. It consists of a sliding resistance coupled to the crankshaft of the engine, and it transmits electrically the position of the piston to the cathode ray tube screen. A device is incorporated in this displacement-type transmitter to allow for the crank connecting-

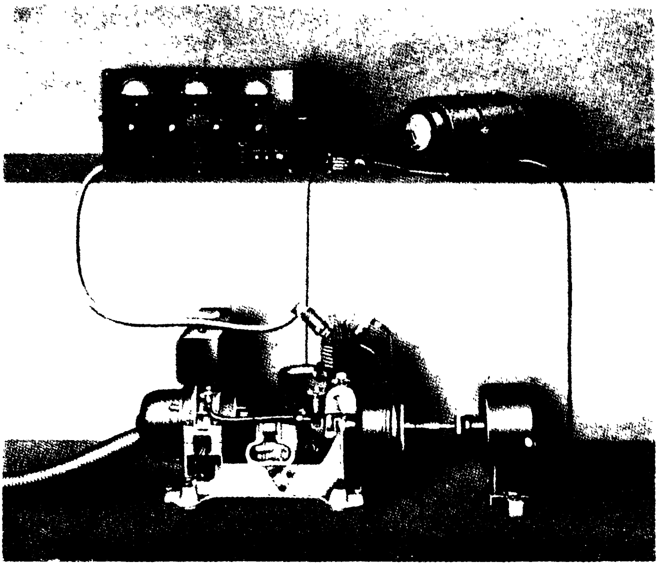


FIG. 269. Zeiss-Ikon piezo-electric indicator.

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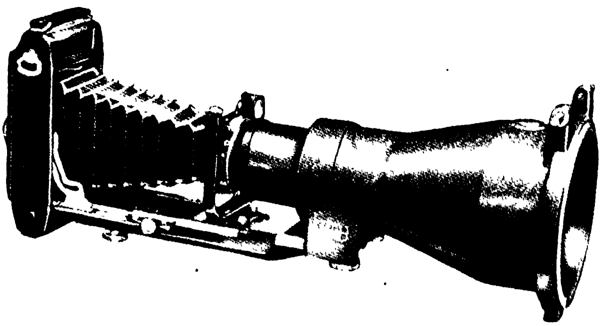

FIG. 271. The Zeiss-Ikon tremograph.

[To face page 307.

rod ratio variation. It is also possible to adjust the phase setting by means of this accessory and thus to produce transposed diagrams.

For time-base diagrams a potentiometer distributor (Fig. 270) is employed. This consists of a central contact sliding over a circular resistance connected across a fixed potential. The contact is driven from the engine crankshaft and is connected to the time-base plates of the cathode ray tube. In this arrangement the potential across the plates is accurately proportional to the position of the engine crank. The pressure element is calibrated statically, by means of compressed air, and in order to simulate engine temperature conditions it is recommended that the pressure unit should be heated before or during calibration.

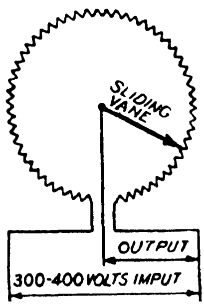

FIG. 270.—Potentiometer of piezo-electric indicator.

Arrangements are made to photograph the pressure-displacement diagrams by attaching a camera to the front of the observation unit as shown in Fig. 271. By the use of another instrument, known as the Tremograph, the pressure-time diagrams can be recorded. This device contains a cathode ray tube which is somewhat different in construction to the one previously mentioned. The movement of the luminous spot corresponding to pressure changes is recorded by a special optical system on photographic paper stretched around a rotating drum. By adjusting the speed of the Tremograph motor, setting the shutter to the exposure time and altering the height of the drum, one or several pressure-time diagrams can be recorded.

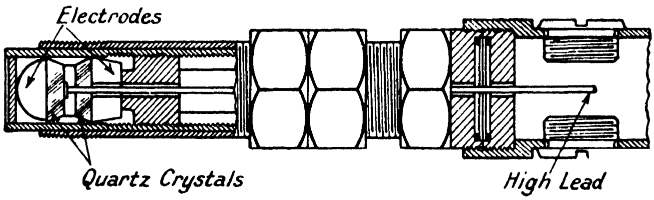

FIG. 272.—Pressure unit of R.C.A. indicator.

The pressure element of the R.C.A. indicator [1] is shown in Fig. 272. It employs two Rochelle salt crystals for ordinary vibration studies as these are about 7000 times more sensitive than quartz in their piezo-electric effect. As these crystals are unsuitable for high temperatures and pressure-shocks they are replaced by quartz ones for internal combustion engine indicators. The two quartz crystals are located in a metal tube between two metal electrodes,

[1] R.C.A. Manufacturing Co. Inc., Camden, N.J., U.S.A., described fully in *Automotive Industries*, March 14, 1936.

the latter being earthed to the shell of the pick-up device, which latter is screwed into a spark-plug hole, or if no extra spark-plug hole is available, into a special hole drilled through the engine head into the compression chamber. The gas pressure within the cylinder is communicated to the lower electrode, and through it to the crystals, by means of a diaphragm of stainless steel which closes the lower end of the tube containing the crystals. The two crystals are placed in the tube in the reverse direction with relation to their electrical axes, so that the adjacent faces, which are separated by a third, insulated electrode, develop a charge of the same sign when the crystals are subjected to pressure. From the third electrode a lead extends through the axis of the pick-up device.

Vibration Tests.—In making ordinary vibration tests (frequency and amplitude), the pick-up device shown is merely held against the vibrating body. The crystals in the pick-up device are supported at three of their corners, and any vibration imparted to the pick-up device is transmitted to the crystals through the mounting surfaces. The mere inertia of the crystals results in deformation of the crystals when they are subject to vibration, and the resulting variations in voltage between opposite faces of the crystals are registered by the oscillograph on the tube screen. The oscillograph can also be used for determining the amplitude and frequency of *torsional vibration*; the method employed is described in the article referred to in the footnote on page 307.

The pressure element of the Kluge and Linckh piezo-electric indicator is shown in Fig. 273.[1] It employs two quartz crystals cut in the form of discs such that the direction of pressure on them coincides with one of the "electrical" axes of the crystals. The steel plug has a very thin bottom which serves as an elastic intermediate member transmitting the gas pressure to the crystals. A measuring electrode is interposed between the two crystals, whose polarities are opposed, so that when pressure is applied the resulting electrical charges of the two combine. Initial pressure is applied to the quartz crystals by means of a screw cap and steel ball. The use of the crystals in opposition obviates the need for insulators, the two outer electrodes being earthed. The measuring wire, which is screened against inductive influences by means of a metal tube, leads to the cathode ray tube or a string galvanometer.

In the case of the pressure element of the Watson and Keys' indicator,[2] shown in Fig. 274, this is combined with the sparking plug. Six quartz crystals are employed and the electric charges are increased proportionately.

The adjacent crystals are reversed with respect to polarity. Metal plates are inserted between the crystals, and all such plates

[1] *Automotive Industries*, May 26, 1934.
[2] H. G. I. Watson and D. A. Keys, McGill University, Montreal.

in contact with negative faces of the crystals are electrically connected to terminal K, while the plates in contact with positive faces are earthed through the hemispherical pressure plate G and the adjusting screw H. By means of the adjusting screw the initial pressure on the crystals is adjusted so that there is no electric charge on the crystals. The charges on the crystals will then vary in direct proportion to the combustion chamber pressure, the charge becoming negative for pressure below atmospheric in the combustion chamber. To reduce leakage, the crystal assembly is shunted with a variable mica condenser (0·001-0·006 microfarad).

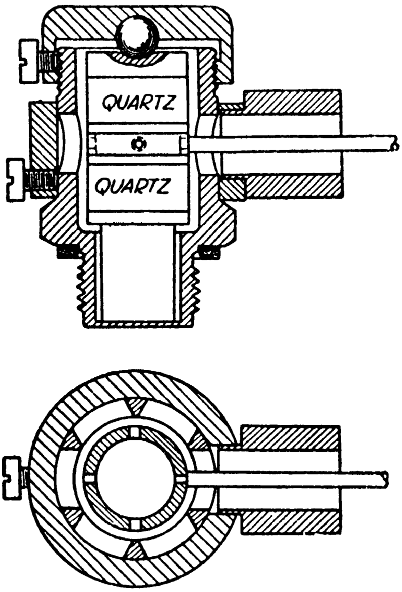

FIG. 273.—Kluge and Linckh pressure unit.

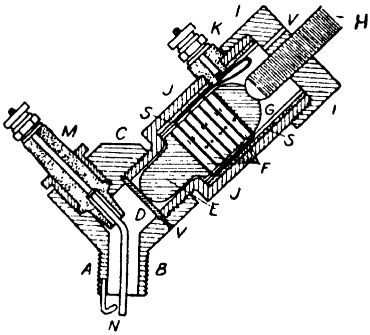

FIG. 271.—The piezo-electric element of Watson and Keys' Indicator.

KEY.—A, B. Threaded portion of adapter screwing into sparking-plug hole. C. Boss on adapter, drilled and tapped to receive pressure element. D. Pressure plate. E. Steel block with hemispherical end transferring gas pressure to quartz discs. F, G. Metal hemisphere transmitting pressure of adjusting screw H to quartz discs. I. Cap for steel tube J containing the quartz discs. K. Binding posts. M. Spark-plug insulator. N. Spark points. S, S. Bakelite tube. V, V. Vents.

The small potential variations across the condenser produced by the crystal detector are amplified and applied to the pressure plates of the cathode ray tube.

Sources of Error in Cathode Ray Indicators.—The principal sources of error to which these indicators may be liable are (1) Inertia effects, (2) Indicator passage effects, (3) Temperature effects, (4) Low resonance frequency, (5) Hysteresis, (6) Electrical errors.

(1) *Inertia Effects.*—As compared with the optical and most other high speed indicators the inertia due to the moving parts of the pressure element is very low in most instances, although in a

few cases the weight moved by the pressure variations is sufficient to introduce errors. In indicators of the Standard Sunbury type a very thin diaphragm is employed, and its inertia effect is almost negligible.

(2) *Indicator Passage Effects.*—The passage between the combustion chamber and the pressure element, if of any appreciable extent, causes *a time-lag* corresponding to the time taken for the sound waves to travel the length of the passage; further there is a tendency to add to the pressure lines on the diagram a vibration equal to the natural frequency of the passage. There is also another effect known as the "sound-transformer" one which occurs when detonation takes place and results in exaggerating the detonation waves. The general result, therefore, is that indicator passage effects give rise to undesirable modifications of the indicator diagram. These should be avoided or reduced to minimum proportions by making the passage as short as possible or avoiding it altogether. In certain engine designs the diaphragm of the pressure element can be exposed direct to the gases in the combustion head without the necessity of a passage, but this is not always possible. Tests should be made with passages of different dimensions to ascertain the effect upon the pressure diagram and thus to obtain an indication of the method of minimising this source of error. In this connection reference has already been made to the special indicator plug used on the Standard Sunbury pressure unit (Fig. 246). The passage wave can, however, be avoided either by an electrical filtration method, a variable band pass filter circuit being employed, but there may be some slight distortion of the pressure diagram on this account.

(3) *Temperature Effects.*—The calibration of the pressure element, according to the type used, may be different when at the normal working temperature to that of the cold element, so that wherever possible the calibration should be carried out in the hot condition; this is more particularly the case with piezo-electric pressure elements.

The thermal stresses in the pressure element are usually much higher on air-cooled than water-cooled engines, so that it is frequently necessary to employ water-cooling for the pressure element when used on the former type of engine.

The piezo-electric indicator has special merits in regard to temperature effects, for tests have shown that the piezo-electric effect of quartz when the temperature is raised from 27° F. to 544° F. varies by only 4 per cent. Whilst this is true for the quartz crystal appreciable errors may occur with complete pressure units owing to the thermal expansion of the crystal housing altering the initial contact pressure between the crystal and its contact members.

(4) *Resonance Frequency.*—For accurate diagrams the natural frequency of the pressure element should be much higher than

any engine frequencies that may occur. For this reason low frequency elements may give rise to resonance effects on the pressure diagrams. In this connection it is here of interest to note the previously mentioned fact of the vibration frequency of the Standard Sunbury indicator, namely, about 40,000 cycles per second—a value well above any engine frequencies.

(5) *Hysteresis Effects.*—If the diaphragm or pressure element (magnetic or electrical units) experiences any hysteresis effect, namely, different pressure readings for the same deflection of the diaphragm when the pressure is increasing or decreasing, then pressure scale errors will be introduced and the indicator diagram will become distorted. It is therefore important to avoid any hysteresis effects, whether mechanical, electrical, or magnetic, in the diaphragm and other pressure element members.

(6) *Electrical Errors.*—These include errors in the electrical amplification of the pressure-potential difference readings and in the time-base conversion methods, the result of which is to render the indicator diagrams worthless for indicated horse-power measurements and misleading for direct observation purposes.

Indicators should have *a relatively long life* under engine running conditions so that new designs should be given a duration run of at least ten hours, after which the calibration should be checked against the initial values and should show no appreciable change. In this connection the carbon-pile pressure element seldom maintains its initial calibration values for appreciable periods of engine running.

Indicators should also be proof against *the effects of mechanical shock*, such as valve shocks, as distinct from cylinder pressure shock. A test, known as the DVL one,[1] consists in dropping balls of 10 and 13 mm. from a given height on to the pressure element; better results are obtained when a cylindrical bar of 10 mm. diameter and 260 mm. length is dropped with its axis vertical. The effect of shocks is usually to cause a shift of the zero line on the diagram and sharp "pressure" peaks on the pressure variation lines, corresponding to the times of the impacts.

Indicator for Compression-Ignition Engine Research.—A new design of three-tube cathode ray indicator has been developed by Prof. R. A. Rose of Wisconsin University, for combustion research on compression-ignition engines; it can also be applied to petrol engines. It enables five important factors connected with combustion to be recorded simultaneously, namely: (*a*) Lift of injection nozzle valve, (*b*) Fuel combustion radiation, (*c*) Cylinder pressures, (*d*) Top dead-centre position, (*e*) Cylinder combustion radiation.

[1] "Valuation and Tests of Electrical Engine Indicators," F. Lichtenberger, *Autom. Industries*, April 15, 1940.

The four chief elements of the Rose indicator system, shown in Fig. 275,[1] are (A) A multiple pick-up and amplifying unit, (B) A three-tube cathode ray oscillograph, (C) A revolving drum camera, and (D) A dead-centre indicator.

Referring to Fig. 275, the fuel injection indicator and amplifier is shown at A-1; the radiation indicator at A-2; the pressure indicator at A-3, and the three-tube cathode ray oscillograph at (B).

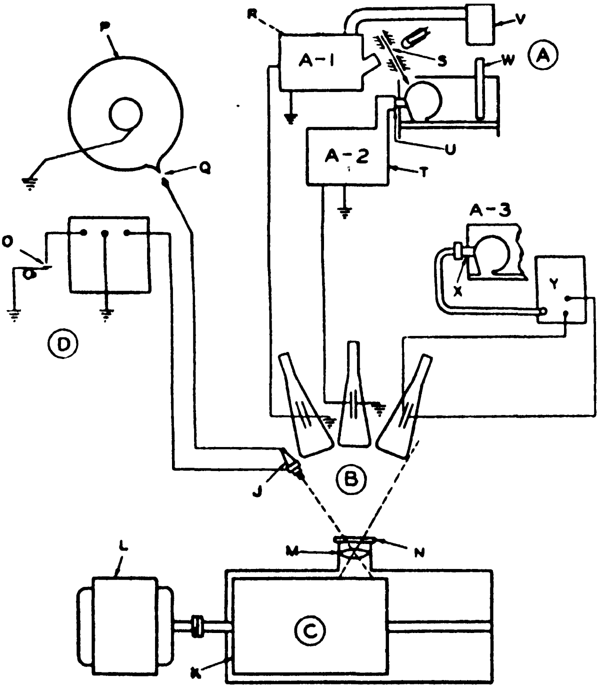

FIG. 275.—Layout of the Rose indicator.

KEY.—J. Sparking plug. K. Sliding film drum 3 ft. circumference carries 20 × 24 in. film. L. Synchronous motor 1200 r.p.m. M. 3 in. f./1·5 lens. N. Shutter. O. Breaker points. P. Engine flywheel—24 in. dia. Q. Rotary air-gap. R. Fuel injection phototube and amplifier. S. Light shutter. T. Radiation phototube and amplifier. U. Separate chamber window. V. Phototube. W. Main chamber window. S. Quartz crystal. Y. R.C.A. amplifier.

The other components indicated by the letters are described below the layout diagram (Fig. 275).

For the purpose of recording the movements of the injector valve, the movable part of a light shutter is fastened directly to the needle valve thereof. A concentrated-filament light source is placed on one side of the shutter and a photo-cell on the other. The photocell is connected to a direct-coupled amplifier, which has an instantaneous output that is directly proportional to the lift of the valve.

[1] *Journ. Soc. Autom. Engrs.*, 1940-41. Reproduced in *The Automobile Engineer*, March, 1941, and *Autom. Industries*, Jan. 15, 1941.

Another photo-cell is energized by the combustion radiation, a fused-quartz window being installed in the side wall of the combustion chamber of the engine directly beneath the fuel injector and on a level with the centre of the chamber. The pressure indicator consists of an R.C.A. quartz-crystal (piezo-electric) pick-up and its amplifier. A separate 5-inch cathode-ray tube is connected to the amplifier of each of the three pick-up systems described, the three screens facing the lens of the camera. On these screens the beams deflect horizontally in the same plane with the axis of the camera drum. The action of the beams on the screens is retained photographically on the film in the revolving-drum camera.

The fourth unit of the system serves to indicate the exact top dead-centre point on the revolving film. On the engine camshaft are mounted breaker points which are adjusted so that a spark coil is energized a few degrees before the top dead-centre. The output from the coil passes through a rectifier valve and then to a small condenser arranged in series with a sparking plug and a rotary air-gap on the engine flywheel. The condenser can only discharge when the rotary air-gap is at its minimum width ; this gap is adjustable so that the minimum position occurs precisely at top dead-centre. The sparking plug is mounted close to the cathode ray tube screens, where its discharge is recorded as a dot on the revolving film.

Photo-electric Cell Indicator.—Photo-electric cells have been employed successfully in high speed indicators, a typical example being that of the Labarthe indicator which has given satisfactory diagrams on two-cycle engines operating at 7000 r.p.m., without any trace of fluctuations on the expansion line.

This indicator employs a pressure diaphragm arranged on the sparking-plug unit. The outer face of the diaphragm, which is made from the high nickel-iron alloy, known as Elinvar, of 0·3125-inch diameter, and 0·040-inch thickness, is polished and chromium plated so as to form a mirror. The principle employed is that of a parallel beam of light which, when reflected from the convex mirror surface of the diaphragm, is spread and therefore reduced in intensity according to the convexity of the diaphragm. The latter reflects the incident beam from a tungsten-filament bulb on to the sensitive surface of a photo-electric cell and the variation of light intensity—which depends upon the convexity of the diaphragm mirror—causes changes of current which are amplified and applied to a cathode-ray oscillograph pressure plate. The indicator has a special device in the form of a rotating circular plate arranged eccentrically in a tube, so as to vary the area of a slot in the latter in direct proportion to the piston's position in its stroke ; this arrangement takes account of the ratio of connecting-rod to crank radius for any given engine. The intensity of a beam of light, incident upon a photo-electric cell, is thus made to vary as the slot area and therefore the piston position.

CHAPTER IX

INDICATOR DIAGRAMS

Piston Displacement Motion.—1. *Connecting-rod to Crank Ratio.*—In the case of petrol engines the ratio of the connecting-rod to crank is fairly small (it varies from about 3·0 to 4·0 in automobile engines). For this reason the piston has not a simple harmonic motion ; the piston moves rather faster during the first part of its inward stroke, and slower during the latter part, than if the connecting-rod were infinite in length. It is necessary, therefore, in the case of indicators to arrange the piston motions to correspond exactly to those of the engine. The ratio of connecting-rod to crank in the case of indicators having a piston displacement motion obtained by means of a connecting-rod crank mechanism, must, therefore, be the same as for the engine itself.

In most cases means are provided for adjusting this ; thus, in the Watson indicator, the ratio mentioned can be altered by varying the radius at which the rocking mirror connecting-rod is attached to the eccentric.

2. *Phase of Piston Motion.*—Not only is it necessary to ensure that the indicator's piston travel motion is an exact replica of that of the engine to be tested, but also that the two motions should be in phase ; the indicator motion should represent that of the main piston in its correct position. Indicators should be provided with means for obtaining, whilst the engine is running, a correct phase adjustment.

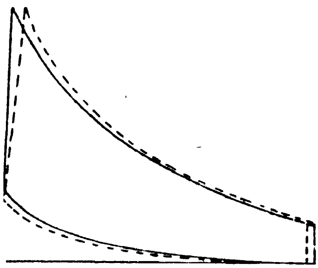

FIG. 276.—Effect upon diagram of advanced phase of indicator.

Where the piston travel of an indicator is derived from the crank or cam shaft of the engine through shafts and gearing, the backlash and torsional strains of the shafting generally cause alterations in the phase, at different engine speeds, so that phase adjustment must be made prior to any test at the actual speed.

If the phase of the indicator is *in advance* of that of the engine, the indicator diagram will be enlarged, thus giving a higher I.M.E.P. value than the real one. This enlargement is shown, diagrammatically, in Fig. 276.

In one example in which a Watson indicator was used by the writer, an alteration in phase of 6° in advance of the correct position raised the I.M.E.P. from 89 to 100 lb. square inch.

Experiments made at the Air Ministry Laboratory[1] upon a Watson-Dalby indicator showed that an error in phase setting of 1° of crank angle at 1200 r.p.m. gave an error in I.M.E.P. of 2 per cent. at full load, and 3 per cent. at half load.

If the phase of the indicator lags behind that of the engine, the indicator diagram will be of smaller area; in this case also a phase

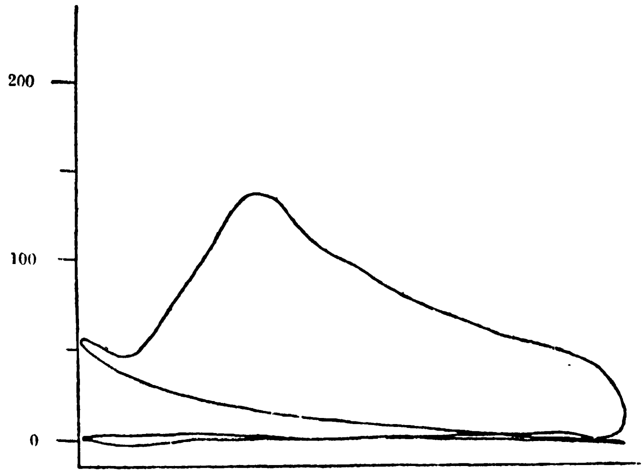

FIG. 277.—Phase in advance.

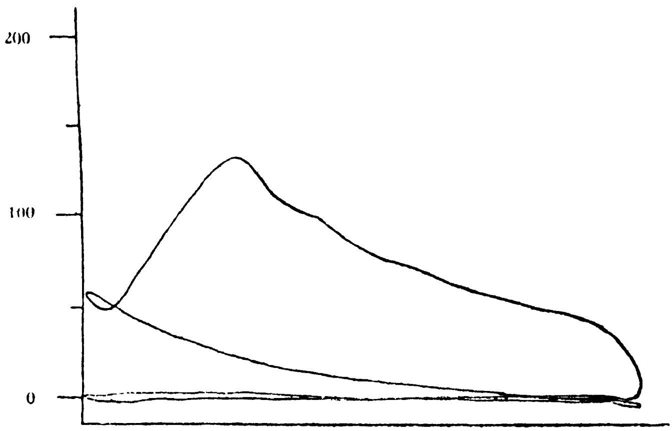

FIG. 278.—Phase lagging.

error of 1° corresponds to between 2 and 3 per cent. error in the I.M.E.P. These values have been verified from theoretical considerations of the indicator diagram, as well as by practical tests upon aircraft engines.

[1] "Phase Setting of Engine Indicators," H. Moss and J. Stern, Air Research Committee Report and Memoranda, No. 878, 1923.

3. *Methods of Setting the Phase.*—In the case of optical indicators, the phase can be set very accurately in a simple manner. The principle of the method will be apparent from Figs. 277, 278, and 279. Fig. 278 shows the effect upon the indicator diagram of a phase advance beyond the correct position, when the engine was running with retarded ignition. In Fig. 278 the ignition is still retarded, but the indicator lags behind the engine. When the engine and indicator are in correct phase, the diagram is as shown in Fig. 279. Hence it is only necessary to run the engine at its given speed, with retarded ignition, and to adjust the indicator's

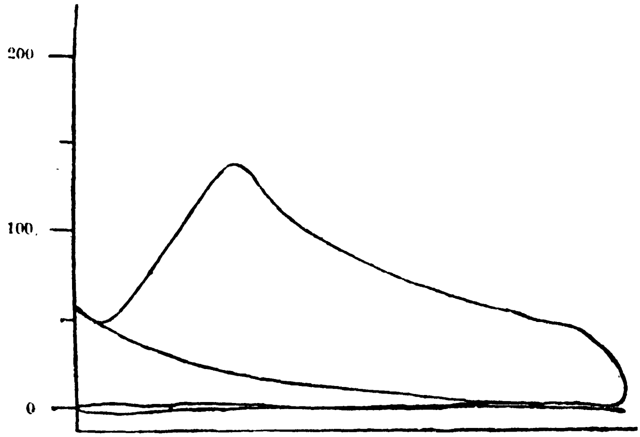

FIG. 279.—Phase correct.

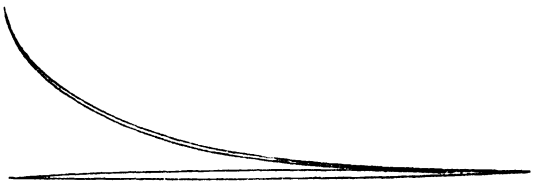

FIG. 280.—Testing phase relation with ignition switched off.

phase until the compression line returns upon itself, prior to the explosion pressure rise.

Alternatively the engine cylinder in question can be temporarily switched off, whilst running, when the last part of the compression line should return on itself (Fig. 280).

In this respect it should be noted that the lower part of the expansion line will be below the compression line, due to the cooling of the gases, and to leakage. It is for this reason that the earlier part, only, of the compression line is taken, for in this case there is practically no time for cooling or leakage to occur. Using the full compression line method, the expansion line will be above the

compression line for advanced, and below it for retarded phase of the indicator at the end of the compression stroke.

It has been shown [1] that the above method for phase setting will, if correctly carried out, be accurate to within $\frac{1}{4}°$, corresponding to an accuracy of $\frac{1}{2}$ per cent. in the I.M.E.P.

4. *Crank-angle or Time Base Diagrams.*—In the case of indicators which produce diagrams upon a time or crank-angle base, the connecting-rod to crank ratio only comes in when it is required to convert the diagrams into p.v. ones. For this purpose it is convenient to have a special celluloid or tracing-paper scale, with a crank-angle base of the same length as that given on the diagrams, and with a series of vertical ordinates drawn through definite piston positions corresponding to the crank-angle ones. This transparent scale can then be laid over any diagram and the values of the ordinates (to any suitable scale, or to the pressure scale if it is known) read off for different piston positions, and transferred to a piston base, for the p.v. diagram. The construction of the transparent scale is quite a simple one. The basis of the method is to draw the line diagram for the piston and connecting-rod positions, using the connecting-rod to crank ratio of the engine in question, and to mark off the piston positions corresponding to definite crank angles. If, now, a horizontal line be taken, of length equal to that of the indicator diagram (crank angle) base, it can be marked off with the definite angle values just mentioned, when the piston positions corresponding to these will be known.

Other Sources of Error in Indicators.—1. *Connecting-tube Bore.*—The bore, and length of the communicating passage between the cylinder head and the indicator piston, or diaphragm, have a marked effect upon the accuracy of the diagrams obtained.

If the bore of the passage is too large, a relatively large volume of hot gases will sweep backwards and forwards, with increased friction and viscosity effect ; these hot gases may also prove detrimental to the pressure unit ; the compression of the engine will also be lowered to some extent.

It will be obvious, then (particularly in the diaphragm and piston types of indicator), that the maximum diameter of the bore of the communicating passage is limited. In any given case, the presence of too large a bore can be examined by taking indicator diagrams, firstly with the indicator cock full open, and secondly with it partly closed, and in one or more positions. If there is any alteration in the general shape, or in the characteristics of the diagram, such as a diminution in the amplitude of the waves shown on the expansion line by partly closing the tap, then the passage bore may be taken to be incorrect. Test indicator diagrams taken with

[1] *Vide* footnote p. 315.

pipes of different bores will at once show whether the bore is too large.

Conversely, too small a bore will cause the gases to be throttled in the passage, and the pressure will be damped down ; this results in lower values of the pressures, as shown on the diagram, and, therefore, of lower I.M.E.P. values than the actual ones.

The only sure way of testing for correct dimensions of bore is to try several of different diameters, and to note the characteristics in the above-mentioned manner.

In the case of the Watson type indicator, the correct diameter of the connecting-tube was found to be from 4·0 to 4·5 millimetres.

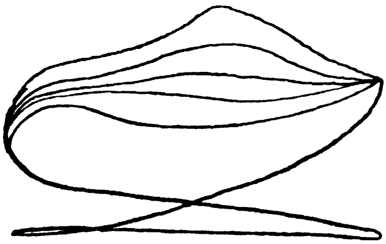

FIG. 281.—Normal short-tube diagram.

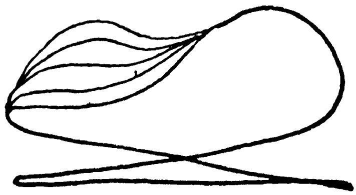

FIG. 282.—Bore wave effect due to long connecting-tube.

2. *Connecting-tube Length.*—The communicating passage between the engine and indicator must be as short as possible ; the shorter it is, the better. It is not possible, always, to realize the ideal in this respect, owing to the fact that a shut-off cock must be interposed between the engine and indicator, and also to the design of the indicator. In the cases of the (Dobbie-McInnes) " Farnboro " and Midgeley indicators, the pressure elements are virtually on the cylinder walls, but in many optical indicators there is an appreciable length of passage.

If the passage is too long, there will be a retardation, or lag, of pressure, and the diagram, being " out-of-phase," will be incorrect. The effect of too long a bore can usually be rectified at any given speed by altering the phase of the indicator, so that a normal type of diagram is obtained when the engine is running normally.

This will, perhaps, be better understood by reference to Figs. 281 and 282, which show a normal and long-tube diagram respectively. Both were taken when the engine was running " light " ; the five expansion curves in each case represent the variations in pressure due to the difficulty of keeping the load, speed, and mixture strength constant under the light load conditions.

In the case of Fig. 281, it will be noticed that the expansion

lines have become shifted towards the "toe" of the diagram; this is due to the lag produced in the pressure attainment, by the long bore of tube.

To consider a concrete case, namely, that of a connecting-tube of $3\frac{1}{2}$ inches length, used on an engine working at 1200 r.p.m. The period of the piston's stroke will be ·025 second.

Now, the maximum rate at which pressure can be transmitted through a gas is identical with the velocity of sound through that gas (sound being a compression wave effect). This rate may be taken as 1100 feet per second for the purpose of this example. The time taken for a pressure wave to travel at the maximum rate through the $3\frac{1}{2}$-inch tube works out at 0·0025 second, or $\frac{1}{100}$ the period of time of the piston stroke. There will thus be a lag amounting to $\frac{1}{100}$ of the stroke in the pressures indicated. Moreover, it will be seen that this lag varies with the length of the tube.

As mentioned before, the effect of this lag can be corrected, within limits, by a phase adjustment.

3. *Resonance Effects.*—The explosion wave, more particularly in cases in which the pressure value is high, or detonation is present, sometimes sets up a resonance, or a pressure fluctuation effect in the indicator, which results in a wavy expansion curve as shown in Fig. 283. These waves must not be confused with those caused by the "period" of the indicator control spring, which are somewhat similar.

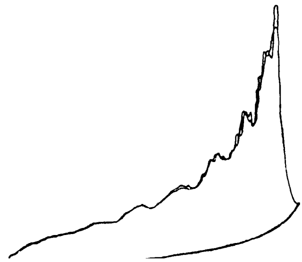

FIG. 283.—Resonance effects revealed by the indicator diagram.

In the present case, these resonance effects frequently have superposed upon them waves of smaller amplitude due to vibration of one of the indicator components.

The effects of resonance can be avoided by the insertion of wire gauze in the communicating tube; this can be arranged to damp out the pressure waves, without, however, throttling the gases.

This method has been employed in the case of a sleeve valve engine, in which the shortest tube possible was 4 inches in length. The diagram obtained at 1000 r.p.m. before the insertion of a copper spiral is that shown in Fig. 283. The insertion of the copper spiral mentioned enabled a smooth expansion line to be obtained.

4. *Spring Period Effects.*—It has previously been stated [1] that if the period of the spring is within the range of speeds of the engine, and the movement of the indicator piston be appreciable, the loaded spring during the expansion stroke (in particular) will vibrate at an appropriate frequency, so that a wavy expansion line results.

[1] *Vide* p. 241.

Many indicators and diagrams suffer from this defect. The only rational methods of correcting this defects are: (1) by employing stiffer springs; and (2) by reducing the amount of movement of the pressure element.

The frequency of a good indicator spring is, as we have seen, at least 30 times that of the engine speed; at these high frequencies, the small amplitude vibrations are effectively damped out by the friction of the other moving parts.

The de Juhasz and Gale indicators have an advantage in this respect, since the pressure recording element moves at only a fraction of the engine speed.

5. *Cylinder Waves.*—Another cause of wavy expansion lines is the vibration effects occurring within the cylinder itself. This is more often the case when the charge is ignited from some extreme point in the cylinder head, and the indicator connection is placed near this point. Fig. 284 is a diagram obtained from a T-headed engine when the charge was fired over the exhaust valve, the engine in this case running at 1100 r.p.m. When, however, the charge was fired over both valves simultaneously, these cylinder waves disappeared, and the M.E.P. at this speed showed an increase of 13 per cent.

6. *Other Sources of Error.*—Other sources of error, most of which have already been mentioned, include leakage of gas past the piston, piston friction, pencil friction, inertia of the parts of the indicator, drum and drum mechanism errors, temperature effects upon the pressure element, wear in the moving parts, vibration of the parts, and other factors.

Applications of Indicator Diagrams.—Provided it is satisfactorily designed, adjusted, and operated, the high speed engine indicator is a most valuable asset for research and ordinary test purposes.

With the optical type, particularly, one is able to follow every phase in the working cycle of an engine under practical conditions, and to ascertain quite definitely the effects of any carburation, ignition, or other adjustments upon the pressures within the cylinder; the indicator, in fact, is the open window of the cylinder. Obviously if the pressure values at various parts of the stroke, or cycle, can be observed directly, and records taken, the engineer will be saved a good deal of reckless speculation and groping about in the dark; the trial-and-error efforts of the past will be superseded by direct observation.

On the other hand, the history of high-speed indicator development has shown clearly that a certain proportion of indicator diagrams taken have been rendered quite worthless owing to a lack, on the operator's part, of a proper knowledge of the principles and facts concerned. It is not a difficult matter to obtain indicator

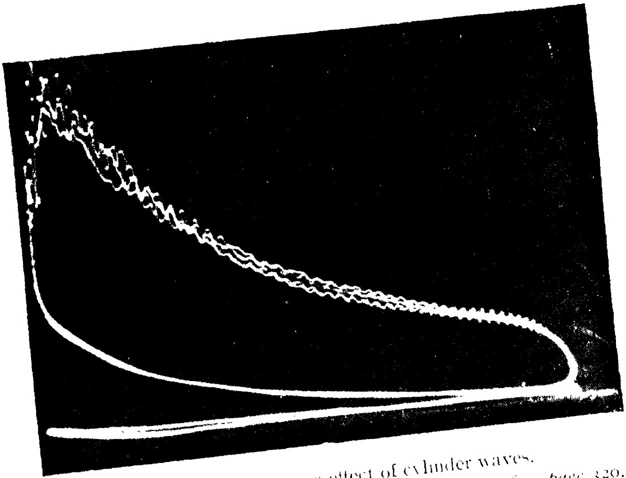

FIG. 281. Showing effect of cylinder waves.

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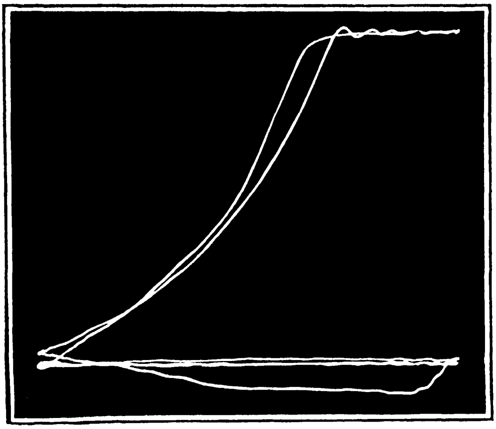


FIG. 286. Light spring diagram from petrol engine taken during motoring tests.

[See page 321.

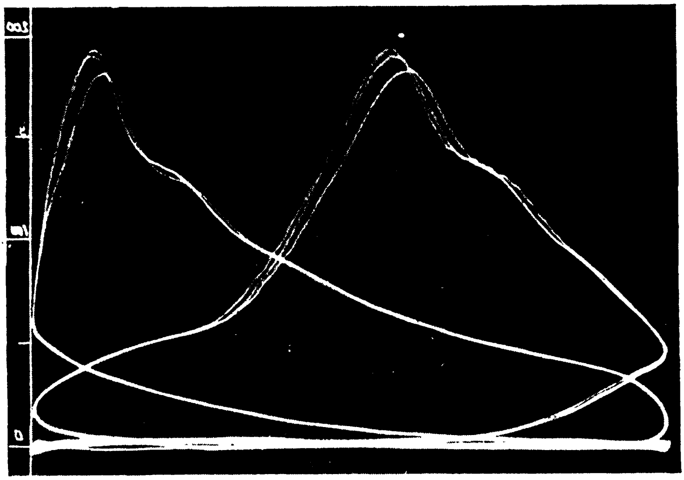


FIG. 287. Normal and 90° displaced diagrams taken consecutively with Watson indicator.

[To face page 321.

diagrams which appear correct, but it is quite a different matter to be able to prove that they *are* correct. It is hoped, however, that acquaintance with the principles and facts set forth in the present chapter will be of assistance in this respect.

Since the indicator is fundamentally a pressure-recording instrument, it can obviously be applied—with suitable precautions—to any cyclical variations of pressure, provided that the cyclical frequency is within its limit.

In the case of the internal combustion engine, the pressure variations in the cylinder itself, in the inlet pipe, exhaust pipe, and in the crank-case, can all be determined.

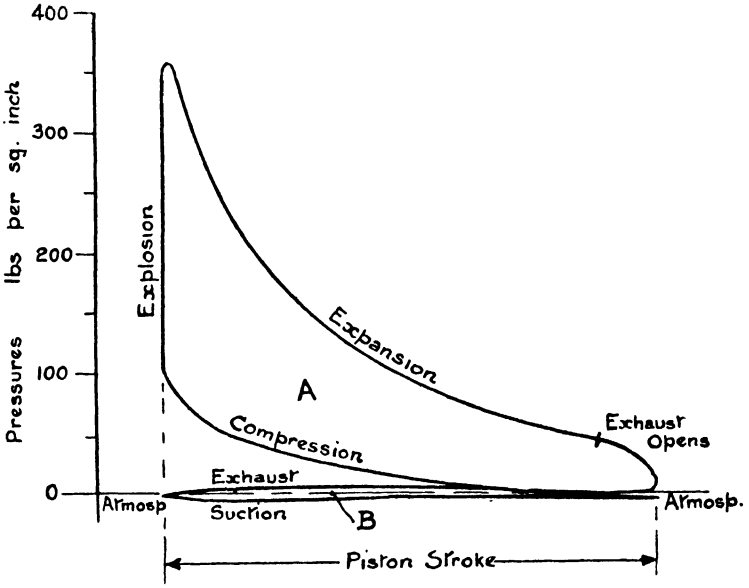


FIG. 285.—Typical indicator diagram.

Indicator diagrams can, in fact, be taken of any pressure variations occurring at engine speeds.

(*a*) *Cylinder Diagrams.*—Fig. 285 illustrates a typical indicator diagram taken from a medium compression petrol engine, and shows the manner in which the pressure varies throughout the stroke.

It will be observed that the inlet and exhaust strokes are so close together that they almost mask each other. For this reason it is usual to employ a special weak spring, provided with a stop above, so that it cannot be deflected by more than about 15 to 25 lb. per square inch pressure.

This enables one to obtain a much more open pressure scale and thus to examine in detail the exhaust and inlet operations.

Fig. 286 is a reproduction of one of a set of optical records,

showing the inlet and exhaust pressures, in the case of an engine which was motored around at 1200 r.p.m. A light spring was also used in this case. The object of the particular tests was to study the relation between the length of inlet piping (between the carburettor and the engine) and the pressure in the cylinder at the commencement of the suction stroke. For this purpose a series of indicator diagrams was taken under each of these conditions, namely, with a different length of inlet piping. It was found, as a result of measurement of the pressures and of examination of the diagrams, that for a constant gas velocity of 96 feet per second, the pressure at the end of the suction stroke in lb. per square inch varied from 14·5 for zero length of inlet pipe, to atmospheric (14·7) for a pipe 1 foot 9 inches long, and thence increased to 15·65 for a 4-foot pipe—the maximum pressure reached, after which it fell progressively with further increase of pipe length. The results indicated the existence of pressure waves in the induction pipe, which, for the 4-foot length of pipe, synchronized with the engine strokes, thus giving a greater pressure in the cylinder than atmospheric. Under these conditions the I.M.E.P. will be greater than the normal value.

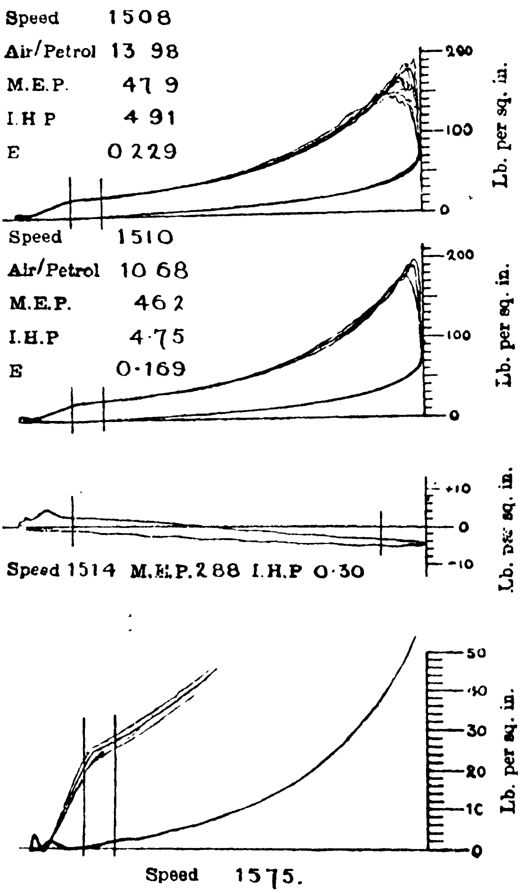


FIG. 288.—Two-cycle engine normal, crank-case and weak spring diagrams.

Fig. 288 illustrates a set of three-port type two-cycle engine indicator diagrams [1] obtained from an optical indicator. The upper one was obtained at 1508 r.p.m., and the second one at 1510 r.p.m., but with a fairly rich mixture (10·68). The corresponding crank-

[1] "Thermal Efficiency of a Two-Cycle Petrol Engine," Watson and Fenning, *Proc. Inst. Autom. Engrs.*, 1910-11.

case pressure diagram, taken with a weak diaphragm, is shown in the third diagram. It will be observed that when the exhaust port is uncovered—as shown by the abrupt drop of pressure at the end of the expansion stroke—there is a pressure rise in the crank-case due to blowback from the cylinder into the transfer port. Thereafter the pressure falls along the crank-case suction stroke following down to minus 5 lb. per square inch, when the piston is at the end of its compression stroke. On the down (firing) stroke the charge in the crank-case is compressed, and about half-way along the stroke reaches atmospheric value.

The lower diagram is a corresponding light spring, one showing in detail the pressure variations during the exhaust port opening period, and when the transfer port is opened.

These diagrams have been included merely for illustration purposes, showing some of the uses to which the indicator can be put.

Displaced Diagrams.—The ordinary indicator diagram is obtained on a stroke base, and the explosion period occurs when the piston is at the end of its stroke, and is, therefore, almost stationary. The resulting pressure line, as may be observed from Fig. 285, is almost a vertical, straight line.

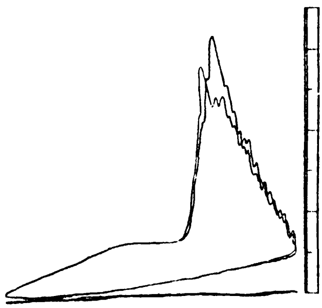


Fig. 289.—Displaced diagram.

If, however, it is desired to study this pressure line in more detail, it must be caused to occur, not when the piston is almost stationary, but when it is moving as quickly as possible.

To obtain this result the phase of the indicator's piston travel motion is purposely displaced so that it is about 90° in advance or retard of the correct position (Fig. 287). The pressure curve during the explosion period thus occurs when the indicator's piston motion is moving at its fastest rate, the result being that the pressure curve is drawn out.

Fig. 289 illustrates the 60° displaced indicator diagram for a sleeve valve engine, the corresponding indicator diagram in correct phase being that shown in Fig. 283. This type of diagram enables one to study pressure rates more accurately than from the correct diagram.

Similarly, if the diagram is arranged on a crank-angle base, as in the (Dobbie-McInnes) "Farnboro" indicator, the velocity is practically uniform, and the explosion part of the pressure curve is extended. The displaced phase and crank-angle diagrams are valuable in connection with ignition, fuel, and detonation tests.

Pumping Loss Diagrams.—It has been mentioned that the mechanical losses of the petrol-type engine are made up of (*a*)

frictional losses due to the rubbing surfaces of the engine, such as the piston and its rings, the cylinder walls, and the bearings ; and (*b*) pumping losses. The pumping losses represent the engine power absorbed in drawing in the fresh charge, and in expelling the burnt gases. The pumping loss in any cycle is represented by the area of the suction-exhaust loop B, Fig. 285. The light spring diagram enables this loss to be measured fairly accurately ; it is usually expressed as a piston pressure. Careful measurements of the pumping losses at various speeds show that in modern engines they are equivalent to a detrimental mean effective pressure which varies from about 2·0 lb. square inch at a gas velocity of 90 ft.-sec. up to 9·5 at a gas velocity of 250 ft.-sec. ; the gas velocity mentioned refers to the inlet pipe.

A study of the light spring pumping loss diagram in conjunction with different inlet and exhaust valve timing settings will usually reveal some important information.

Some interesting examples of inlet and exhaust pipe diagrams are given in the papers mentioned in the footnote.[1] In each case the presence of pressure waves, usually which increase in amplitude with increased speed, is revealed.

Measurements of Indicator Diagrams.—With most types of engine indicator, the resulting diagram is either a pressure-volume one or a pressure-crank angle one capable of conversion to the former.

The net area of this diagram is proportional to the work done per cycle of the engine, and is, therefore, a measure of the indicated horse-power, if the speed is known.

The general type of four-cycle internal combustion engine diagram which is usually obtained is similar in characteristics to that shown in Fig. 285. The area A represents the useful and the exhaust-suction loop B the wasteful work. The effective work per cycle is, therefore, represented by the difference in areas, namely A — B.

It is not necessary, however, to measure the area of the p.v. diagram, but only the mean height, which is proportional to the I.M.E.P., although, if the area can be measured conveniently, as, for example, with a planimeter, the mean height *h* is given by

$$h = \frac{\text{Area in square inches}}{\text{Length of stroke (on diagram) in inches}} \text{ inches.}$$

Then *h*, multiplied by the pressure scale (1 in. $\equiv m$ lb. sq. in., say), gives the I.M.E.P.

The usual method adopted is to divide the indicator diagram into ten equal parts, as shown in Fig. 290, by ordinates parallel to

[1] " The Thermal and Combustion Efficiency of a Four-Cylinder Petrol Motor," Professor W. Watson, *Proc. Inst. Autom. Engrs.*, 1908-9. " Test of a Daimler Sleeve Valve Engine," Professor W. Watson, *ibid.*, 1912-13.

the pressure axis, and to draw the dotted ordinates shown, such that they lie midway between the full line ones.

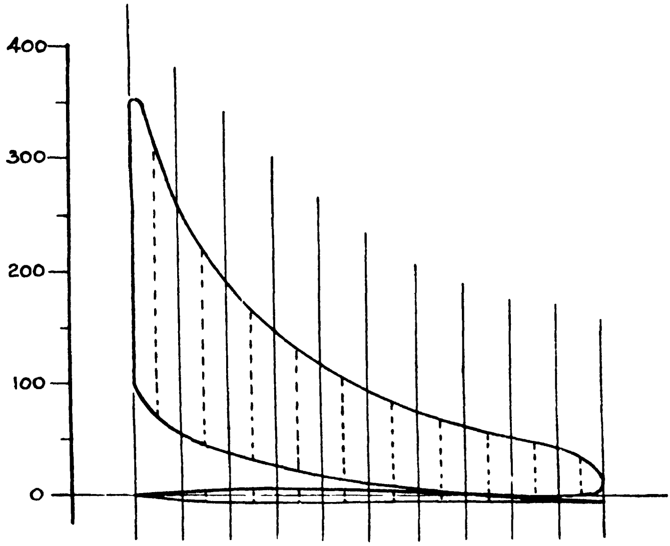


FIG. 290.

The sum of the lengths of the dotted ordinates divided by 10 gives the mean height of the diagram to a degree of accuracy sufficient for most purposes. The method is, of course, equivalent to dividing the diagram into ten areas, each individual area being taken as that of a rectangle of height equal to that of the dotted ordinate; the greater the number of equidistant ordinates taken, the more accurate will be the result.

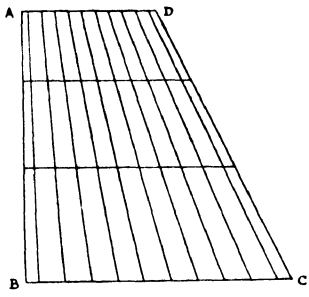


FIG. 291.—Radial divider.

Messrs. (Dobbie-McInnes) "Farnboro" supply a radial divider (Fig. 291) for marking off the dotted ordinate positions quickly. All that is necessary is to mark off and draw the extreme ordinates (corresponding to the stroke ends) and the atmospheric line. The scale is then placed with its edge AB along one side of the extreme ordinates, and is slid up or down this line until the line CD passes through the foot of the other ordinate, when the positions of the mid-ordinates can be plotted or pricked off on the diagrams.

If p_m = I.M.E.P. in lb. sq. in.,
l = length of stroke in feet.,
a = piston area in sq. in.,
n = number of cylinders.,
N = r.p.m.,

then indicated horse-power $= \dfrac{p_m \cdot l \cdot a \cdot N \cdot n}{66,000}$ (4-cycle engine).

The planimeter is a convenient instrument for measuring the area of the diagram directly. Fig. 292 illustrates the Amsler Polar planimeter, which can be used to measure areas directly, the result being given in square inches. It has a range of circle of 18 inches diameter.

It is not necessary to go into the theory of this integrating

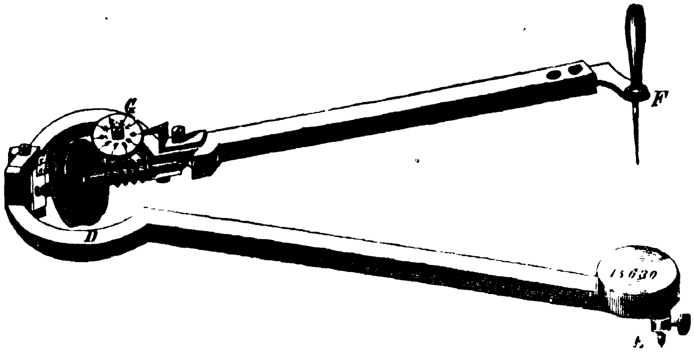


FIG. 292.—Amsler planimeter.

device, which is ably described in other treatises, but rather to describe its method of use.

There is a point E on the instrument which is pricked into the board on which the diagram to be measured is placed securely. A pointer F follows the contour of the diagram (Fig. 292), and the vernier scale reading on the roller wheel D, together with the reading on the worm-driven dial G, gives the diagram area reading. This reading multiplied by pressure scale (1 inch $\equiv m$ lb. sq. in., say), and divided by the length (stroke) of the diagram, gives the I.M.E.P. value.

CHAPTER X

TEMPERATURE MEASUREMENTS

IT frequently becomes necessary in connection with engine tests and research work to know the temperatures of different items of the engine. These include: (1) the cooling water; (2) the exhaust gases; (3) the valves; (4) the piston; (5) the cylinder walls and head; (6) the mixture or gases within the cylinder; (7) the induction system.

For ordinary routine test work, a knowledge of the cooling water temperature is all that is necessary, although the petrol and lubricating oil temperatures should be known.

In cases in which new designs of engines, more particularly aircraft and air-cooled automobile engines, are under investigation, and in connection with fuel tests, exhaust pipe, and silencer design, it becomes necessary to ascertain the temperatures of other items, such as those enumerated above. It may be inquired whether these temperatures cannot be computed, in several cases, from a knowledge of the fuel's physical properties, air-to-fuel ratio, and the compression. In answer to this question it may be stated that any theoretical estimate of the temperatures existing within the cylinder, derived from considerations of the heat cycle, etc., can only approximate, since there are many uncertain factors involved.

For example, the calculated temperature of explosion from the heat cycle is considerably higher, as a rule, than that which has been shown to exist, due to specific heat variations, dissociation wall-action, after-burning or other causes.

On the other hand, certain empirical relations have been formulated as a result of accumulated experience and the results of actual measurements, which, in conjunction with a knowledge of the thermodynamical relations, enable the temperatures of certain items, such as the explosion, exhaust, compression, and suction temperatures to be ascertained with a fair degree of accuracy. It is not proposed to enter into the question of these theoretical and empirical relationships here, but rather to describe the practical methods of ascertaining temperatures in the high speed internal combustion engine.

The reader is referred to the original source of information concerning internal combustion engine temperatures to the footnote references[1] on this and following pages.

[1] (*a*) Callendar and Dalby's "Petrol Engine Temperature Measurements," *Proc. Roy. Soc.*, 1907.
(*b*) Professor Coker's "Gas Engine Measurements," *Proc. Inst. Civil Engrs.*, 1913-14.

Magnitudes of Temperatures.—Before any specific engine temperature can be measured, it is necessary to know, approximately, what is the range of temperatures involved, so that a few remarks on this subject may not be out of place here.

1. *The Cooling Water.*—The maximum temperature concerned is that of the boiling-point of water, namely, 212° F., or 100° C. The average working temperature [1] in modern engines varies from about 75° C. to 85° C. (167° to 185° F.), although some engines consistently run hotter, and some cooler. Working temperatures measured on automobile engines in service indicate temperatures as high as 97° C. (207° F.), and as low as 55° C. (131° F.). A thermometer with a range of 50° to 100° C. (122° to 212° F.) will be satisfactory for cooling water measurements.

2. *Exhaust Gases.*—The temperature of the exhaust gases depends upon several factors, the chief of which are the nature of the fuel, the mixture strength, valve timing, and compression ratio. The design of the combustion chamber, and of the exhaust valve and its port, and the silencing arrangement, also have an appreciable influence. The results of tests made by the author on a four-cylinder car engine running at 1100 r.p.m. (Fig. 293), showed that the maximum exhaust gas temperature, measured in the exhaust manifold near the valves, was 760° C. for a mixture ratio of petrol-to-air of about 15·0 (this ratio corresponded to that of the most complete combustion of the fuel). For mixtures of about 14 and 16, the temperature was 748° C., whilst for mixture ratios of 12 and 18, it fell to about 700° C. The temperature for mixtures of 10 and 20 was found to be 650° C. The results of some gas-engine exhaust gas temperature measurements made by Professor Coker, showed that at the commencement of the exhaust stroke the temperature was about 850° C., falling at the end of the stroke to 600° C. These measurements are applicable to the gases in the cylinder.

The exhaust gas temperature will be lower the farther it is measured from the engine, and higher as the exhaust back pressure increases.

(*c*) " Exhaust Valve and Cylinder Head Temperatures in High Speed Petrol Engines," Professor A. H. Gibson, and H. W. Baker, *Proc. Inst. Mech. Engrs.*, Dec. 1923.

(*d*) " The Air Cooling of Petrol Engines," Professor A. H. Gibson, *Proc. Inst. Autom. Engrs.*, 1919-20.

(*e*) " The Thermal and Combustion Efficiency of a Four-Cylinder Petrol Motor," Professor W. Watson, *ibid.*, 1908-9.

(*f*) " Temperatures in Internal Combustion Engines," Chapter vii, ***Automobile and Aircraft Engines***, A. W. Judge (Sir Isaac Pitman & Sons, London).

(*g*) " Piston Temperatures in a High Speed Air-Cooled Petrol Engine," H. W. Baker, *Proc. Inst. Autom. Engrs.*, 1935-6.

(*h*) " Piston Temperatures and Heat Flow in Petrol Engines," Professor A. H. Gibson, *Proc. Inst. Mech. Engrs.*, 1926-7.

(*j*) " Cylinder Temperature," M. O. Teetor, *S.A.E. Journ.*, Aug. 1936.

[1] Usually taken as the cylinder jacket outlet water temperature value.

3. *Exhaust and Inlet Valves.*—Professor Hopkinson measured the temperature of the centre of the exhaust valve of a Crossley gas engine running with a rich mixture and with the jacket water boiling; he found it to be about 530° C. The corresponding inlet valve head temperature was 330° C. The temperature of pre-ignition was found to be a little over 700° C. in this case. Professor Coker, in connection with some gas-engine temperature measurements, found the average exhaust valve head temperature to be about 400° C. The corresponding inlet valve temperature was 300° C. These values agree with Hopkinson's for a gas engine running under medium conditions, in which the respective average exhaust and inlet valve temperatures were 400° C. and 250° C., accompanied by a cyclical variation of about 15° C.

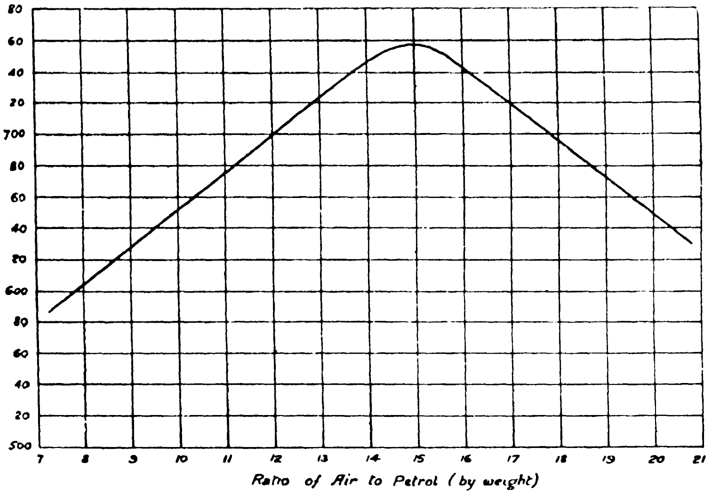


FIG. 293.—Exhaust gas temperatures.

Dr. Gibson's researches on air-cooled aircraft engines indicate that the exhaust valve head temperature varies from a maximum at about 750° C. for an air-petrol ratio of 15·0 down to 700° C. for one of about 11·0, a result which is in general conformity with the author's. Obviously these temperatures will depend upon conditions of mixture strength, nature of fuel, throttle opening, spark advance, and engine speed and design, but the above values may be taken as average results under full-load conditions. Under normal conditions of operation, the temperature of the exhaust valve in modern water-cooled engines may be taken as lying between 600° C. and 750° C.

4. *The Piston.*—The temperature of the piston crown will also depend upon the factors above-mentioned. Further, it will vary with the design of the piston, the material (i.e. whether cast-iron

or aluminium alloy), the clearance between the piston and cylinder, and the lubrication supply.

Professor Hopkinson measured the temperatures of a gas engine piston, and found that for a flat, cast-iron piston face of uniform thickness, the temperature at the centre was about 520° C. Professor Coker, in the case previously cited, found the maximum "mean" piston head temperature at the centre to be 340° C.

The results of Dr. Gibson's researches on aircraft engines, indicated that with air-cooled engines the maximum piston "mean" temperature was between 215° and 250° C.

In the case of water-cooled aircraft engines, with cast-iron pistons, the temperature may be from 120° C. to 180° C. higher than for aluminium-alloy pistons, on account of the inferior thermal conductivity of the former. Thus, in the case of a certain engine, the maximum temperature of the aluminium-alloy piston was 240° C., whilst, when a cast-iron piston was used, it was 400° C., that of the cylinder liner being 145° C.

5. *The Cylinder Walls and Head.*—The early experiments of Professor H. Callendar, in 1904, on a small, air-cooled petrol engine, which was cooled on the exhaust-valve side by means of a fan, showed that the temperature of the combustion head never rose above 200° C. on the exposed side, and 260° C. on the sheltered side. The cylinder barrel "mean" temperature ranged from 150° C. to 200° C. When the temperature of the combustion head reached 300° C. "overheating" troubles occurred. He found that the temperature of the cylinder barrel was only about 20° C. hotter at the top than at the bottom, due to the "piston convection" action.

Dr. Gibson's earlier research work upon aircraft engines showed that with an air-cooled cast-iron cylinder of 100 mm. bore and 140 mm. stroke, the maximum and minimum cylinder head temperatures measured were 268° C. and 171° C. respectively. In the case of an aluminium-alloy cylinder of similar dimensions the maximum and minimum temperatures were 175° C. and 129° C. respectively, under similar conditions. The maximum cylinder liner temperature, in this case, was 144° C. The velocity of the cooling air was 58 m.p.h., and its temperature 15° C. The compression ratio was 4·7, and maximum b.m.e.p. 115 lb. per square inch. The temperature of the liner was, in parts, as much as 26° C. hotter than the cylinder, whilst the maximum temperature recorded was 233° C. on the leeward side of the cylinder near the top of the cylinder head. The maximum temperature difference between the front and the back of the cylinder was found to average about 60° C., whilst the maximum difference between the top and the bottom was 68° C.

As a result of a number of tests, it has been shown that the

maximum cylinder head temperature in the cases above-mentioned should not exceed 280° C. to 300° C. If temperatures exceed about 300° C., the cylinders are apt to distort, valve-seatings and pistons crack, and pre-ignition occur.

The cylinder temperature has been shown to increase with the engine speed. Thus, in the case of some tests made by Dr. Gibson on an aluminium air-cooled cylinder, the temperature on the side of the combustion space rose from 100° C. at 800 r.p.m. to 138° C. at 1800 r.p.m.

The cylinder temperature also depends upon the compression ratio, there being a certain ratio which gives a minimum wall temperature. Thus, in the case of an aluminium air-cooled cylinder of 100 mm. bore and 140 mm. stroke, the mean values of the cylinder barrel temperatures at the top, for compression ratios of 4·6, 5·0, 5·4, 5·8, 6·2, and 6·4, were 180°, 170°, 157°, 154°, 183°, and 212° C. respectively. At the two highest compressions pre-ignition and knocking occurred.

In the case of a single steel, air-cooled cylinder,[1] 5½-inch bore by 6½-inch stroke in an air blast of 83 m.p.h., the cylinder head temperatures at 1450 to 1650 r.p.m. varied from 254° C. to 290° C.

The effect of water-jacket temperature in the case of an Armstrong-Siddeley 30 h.p. water-cooled, overhead-valve engine, showed that the exhaust-valve temperature increased uniformly with that of the jacket water; thus it was 755° C. for a jacket water outlet temperature of 43° C. and 794° C. for one of 88° C.

The cylinder head temperature, however, increased only by about 10° C., namely, from 190° C. to 200° C., for a water-jacket temperature range of 60° C. to 95° C.

It is not possible to devote more space here to this subject, but the reader will find much useful information in Dr. Gibson's original papers, mentioned in the footnote on page 328.

6. *The Gaseous Mixture.*—The temperatures hitherto mentioned, with the exception of the cooling water and exhaust gases, refer to those of the metal components of the engine. These temperatures are necessarily much lower than those of the exploding and expanding gases.

The manner in which the temperature of the gases in the cylinder vary throughout a cycle is indicated in Fig. 294, which refers to a gas engine. It is very similar (but with different individual temperatures) in the case of the petrol-type engine; this can also be shown from theoretical considerations.

The metal of the cylinder walls, piston, and valves is thus subjected to a cyclical variation of temperature, of high frequency, namely, 1000 per minute, in the case of an engine running at 2000

[1] See p. 328 (*d*).

r.p.m., and of a temperature range varying by about 1200° C. to 1800° C.

The lowest gaseous temperature occurs during the suction stroke and, in the case of a petrol engine, varies from about 100° C. to 160° C. The lower value corresponds to an engine running, partly throttled, with low jacket water temperature and at a fairly low speed. The higher value corresponds to full-load and speed conditions, the incoming charge being heated by the residual gases and the metal of the cylinder, valves, and piston to a much greater extent.

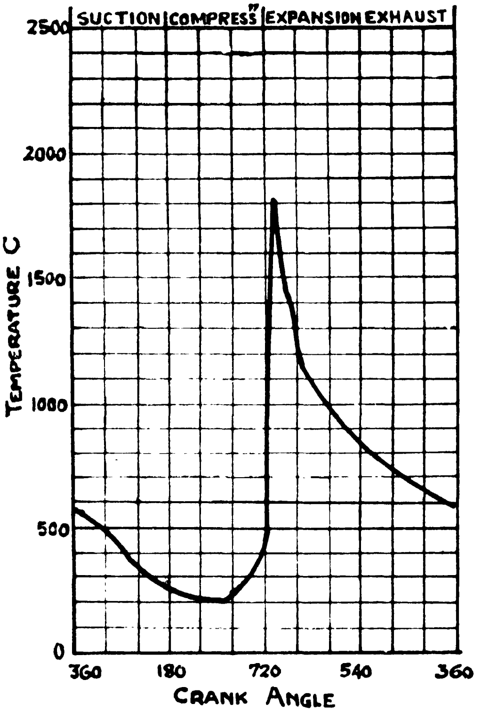


Fig. 294.—Temperature variations during four-cycle period.

The maximum temperature attained during explosion depends upon many factors, including the mixture ratio, compression, volumetric and thermal efficiencies, design of the engine, the ignition engine speed, and throttle opening.

No definite values can, therefore, be given, but the results of certain deductions and direct measurements show that in the case of a petrol engine the maximum full-load temperature lies between the limits 1800° C. and 2500° C.

Professor Coker, from measurements made in a gas-engine cylinder, using thermo-couples, found that the explosion temperature with weak charges was about 1700° C., and a maximum temperature of 1840° C. for an air-gas ratio of 7·35 to 1.

From measurements of suction and compression temperatures in a gas-engine cylinder, Callendar and Dalby deduced the explosion temperature to be about 2500° C. with a rich mixture and 2250° C. for a less rich one of 7·1 air to 1 gas.

Watson gives a maximum temperature value for an air-benzole mixture of 12·5 of 2100° C., and for an air-petrol mixture of 14·4 of 2100° C. also. Just before the exhaust valve opens, the temperature of the gases fell to about 1500° C.

Measurement of Water-jacket Temperatures.—Since the

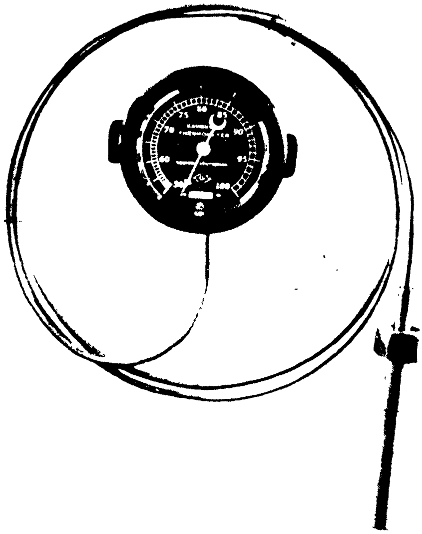


FIG. 205. Aircraft radiator type distant reading thermometer.

[*To face page* 333.

temperature range is only about 50° to 60° C., and the maximum possible temperature 100° C., an ordinary mercurial type thermometer is applicable. The exposed glass-bulb type is rather fragile for routine work, so that the addition of a thin perforated brass shield around the bulb is an advantage. In some cases it is better to insert a thin metal thimble in the inlet or outlet pipe, to partly fill this with mercury, and to insert an ordinary mercury glass thermometer into this mercury bath; this arrangement enables the thermometer to assume the working temperature very quickly.

In many cases it is inconvenient for the test assistant to read the inlet and outlet water temperatures directly, but to employ distant reading instruments, which can then be arranged conveniently to his other apparatus. The ether (or similar volatile liquid) vapour types of distant reading thermometer are particularly suited to this purpose. The aeroplane engine radiator type illustrated in Fig. 295 [1] is quite accurate enough for most test purposes, and the lag is very small. The bulb and tubing containing the liquid or its vapour are usually made of nickel-plated copper, the tubing being sufficiently flexible to bend to any convenient shape for fitting purposes. The dial in the example shown is of 2 inches diameter, and is graduated from 50° to 100° C. These instruments are robust in construction, reliable over long periods, and the needle of the indicator is unaffected by any ordinary engine vibrations. The thermometers can also be fitted with an electric alarm attachment, or relay switch to ring a bell, or to operate another piece of apparatus such as a fan, throttle, etc., when the temperature reaches a certain pre-arranged value.

In all cases the thermometer bulbs should be so arranged in the water system that they measure the mean water temperatures required.

The type of instrument shown in Fig. 295 can also be obtained, with mercury inside the bulb,[2] so as to read from 0° to 500° C., and fitted with either 4-inch, 8-inch, or 13-inch dials. Further, they are also supplied as self-contained thermographs, giving a continuous polar diagram record of temperature fluctuations on a time peripheral scale.

Measurement of Suction and Explosion Temperatures.—A knowledge of the temperature of the fresh charge drawn into the cylinder, at the end of the suction stroke, is important in internal combustion engine research work, as it enables the temperatures and pressures at other parts of the cycle to be computed, and the ideal and comparison diagrams to be constructed.

The fresh charge is heated in its passage through the inlet pipe

[1] Messrs. The Cambridge Instrument Co., Ltd., London.
[2] *Vide* Fig. 314, p. 353.

and valve passages, and also by the cylinder walls and the piston head. Further, it is mixed with a certain quantity of residual hot exhaust gases, so that its final suction temperature cannot, in the ordinary way, be deduced from its initial temperature, say, at the carburettor.

It is possible, however, to deduce the value of the suction temperature from that of the exhaust gases in the cylinder ; this latter temperature can now be measured with a fair degree of certainty and accuracy, either by means of a thermo-couple or electric-resistance type thermometer placed under, but near, the exhaust valve.

If the temperature of the exhaust gases be T° C., and that of the atmosphere t° C., and if V denote the volume of the combustion chamber, then

Volume of fresh charge at atmospheric pressure and temperature t° C. $= (n - 1)V$, n being the compression ratio.

If the density and the specific heats of the gases remain constant, we have

$$\frac{nV}{T_s} = \frac{(n-1)V}{t} + \frac{V}{T} \text{ by Charles's law}$$

where T_s is the suction temperature.

From which
$$T_s = \frac{n \,.\, t \,.\, T}{(n-1)T + t}.$$

This method does not, however, take into account the volumetric efficiency of the engine, nor the increased volume of the burnt gases due to combustion.

It is possible,[1] however, to obtain the suction temperature with a higher degree of accuracy from a knowledge of the cylinder pressure P_1 when the inlet valve closes, the atmospheric pressure P_o, and the true compression ratio n, from the following rational formula :—

$$T_s = \frac{T_o \,.\, T \,.\, n}{T_o \,.\, \frac{P_o}{P_1} + T \,.\, \left(n - \frac{P_o}{P_1}\right)}$$

where T_o = absolute temperature of the inlet charge before admixture with the residuals,

T = exhaust gas temperature (absolute).

All of the quantities on the right-hand side of the equation are known or are determinable. The value P_1 can be obtained from the light spring indicator diagram.

[1] "Note on the Estimation of Suction and Compression Temperatures in Internal Combustion Engines," W. Morgan, *Proc. Inst. Autom. Engrs.*, April, 1923.

The temperature of the charge or compression T_c is given by

$$T_c = \frac{T_s \cdot P_c}{n \cdot P_1} \text{ very approximately}$$

where P_c = measured compression pressure.

The rational formula for the relationship is as follows :—

$$T_o = \frac{T_a \cdot P_1 \left\{ n - \left(\frac{P_a}{P_1} \right)^{\frac{1}{y}} \right\}}{P_a \cdot \eta \cdot (R - 1)}$$

where

T_a = absolute temperature of atmosphere,
η = volumetric efficiency,
R = nominal compression ratio.

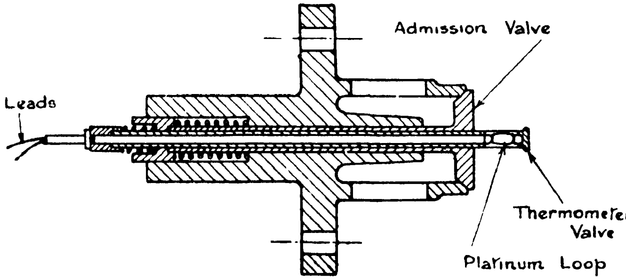


FIG. 296.—Thermometer valve used by Callendar and Dalby.

The approximate formula, however, gives results within about 0·25 per cent., and its employment saves a considerable amount of work.

The value of y, the index value in the adiabatic equation PV_y = constant, may be taken as 1·33.

The suction temperature can be measured direct in the case of slow-running engines ; this has been done by Callendar and Dalby,[1] and later by Coker and Scoble.[2]

The former employed a platinum resistance thermometer, consisting of a fine platinum loop of 0·001 inch diameter (so as to have an extremely low thermal capacity), which was inserted in the stem of the hollow admission valve. By means of a separate cam this " thermometer valve " could be pushed into the combustion chamber, and thus exposed during the inlet (or compression) stroke ; during the explosion and exhaust strokes, when the temperatures were high enough to fuse the platinum wire, the thermometer valve was withdrawn into the stem of the admission valve. By means of an ingenious contact device, the temperature could be ascertained

[1] *Proc. Roy. Soc.*, 1907. [2] *Proc. Inst. C.E.*, vol. 196, p. 1.

at any part of the suction or compression stroke, in a somewhat similar manner to the " pressure sampling " device.

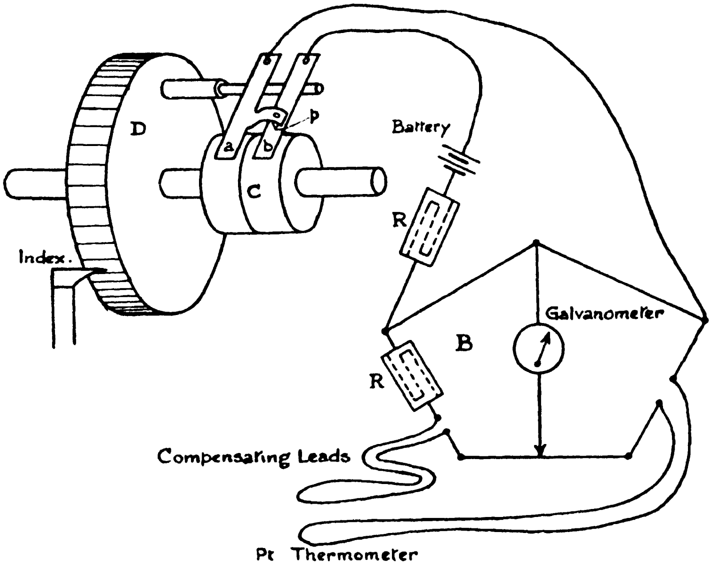


FIG. 297.—Arrangement of Callendar and Dalby's apparatus.

This method was applied to a gas engine running at 130 r.p.m., and even then the peak temperature recorded by the valve thermometer was below the true value ; further, the time-lag of record after reception of heat was about 0·6 second for a temperature variation of 200° C. in half a revolution.

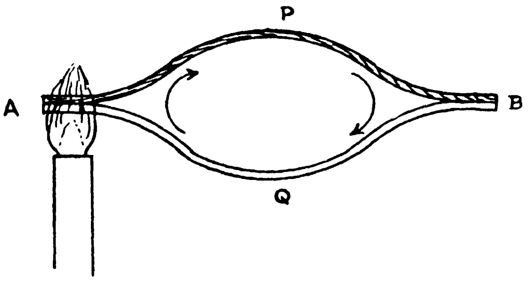


FIG. 298.—Illustrating principle of the thermo-couple.

It is evident, then, that this method cannot be applied without considerable difficulty to high speed engines of the type under consideration.

Thermo-couples.—The principle of this type of temperature-measuring device, which is universally employed for ascertaining temperatures *at a point*, depends upon the fact that when two pieces of wire of different materials, P and Q (Fig. 298), are fused or soldered together at their ends, A and B, and one end, A, is heated, an electric current is generated which will flow around the circuit. The direction of this current will depend upon the materials used ; if

the two wires, P and Q, are of copper and iron respectively, the direction of the current will be as shown.

The existence of a current, in the above case, indicates that there must be a difference of electric potential between the two metals. If t_a and t_b are the temperatures at A and B respectively, then the electromotive force E is given by

$$E = p(t_a - t_b)$$

where p is a constant for the pair of metals used. The E.M.F. obtained by heating the junction is only a few millivolts[1] for

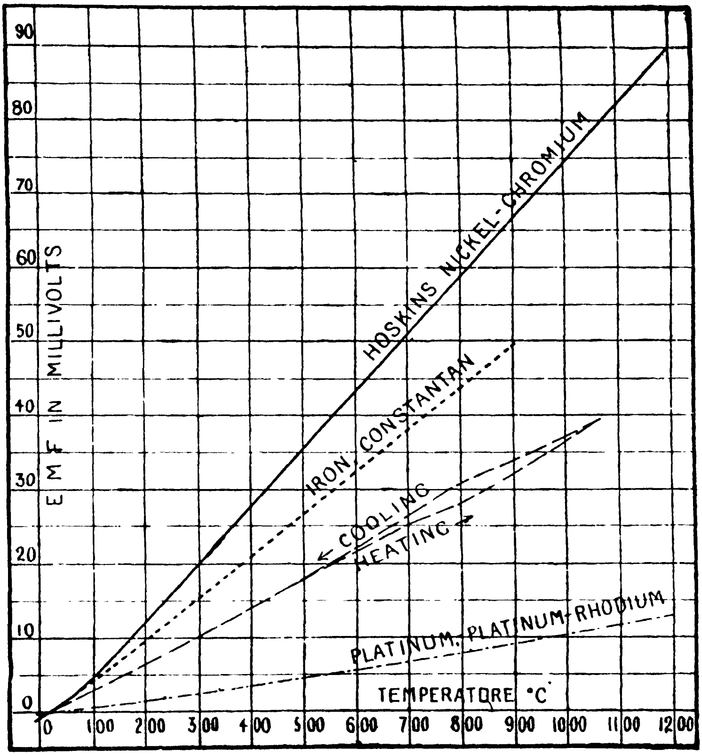


FIG. 299.

100° C. difference of temperature between the heated and the cool junction. It is independent of the size or area of contact of the metal wires or rods.

Fig. 299[2] shows the curves of E.M.F. and temperature differences for a few typical materials. The temperatures denoted in the abscissæ are those of the hot junctions, the cold junction being at 0° C.

[1] One millivolt = $\frac{1}{1000}$ volt.

[2] "Electric Measuring Instruments," D. J. Bolton (Chapman & Hall, Ltd.).

The formula given above holds within certain limits for most simple metals, but a more accurate formula is

$$E = -\alpha + \beta(t_a - t_b) + \gamma(t_a - t_b)^2$$

where α, β, and γ are constants for the given metals.

The method of employing thermo-couples is illustrated, diagrammatically, in Fig. 300. The cold junction is usually obtained by immersing the fused or soldered pair in melting ice, contained in a vacuum flask.

It has been stated that for a given pair of metals, the current will flow in a definite direction if one junction is at a higher temperature than the other. This is true in general for a limited range of mean temperatures above this range, the thermo-electric effect may cease altogether at a certain temperature, and at a still higher temperature it may occur again, but in the reverse direction. Thus, in the case of copper and iron (Fig. 298), if A is at 100° C. and B at 0° C., the current circulates as shown. If, however, A is at 380° C. and B is at 280° C., there will be no current flowing. If, however,

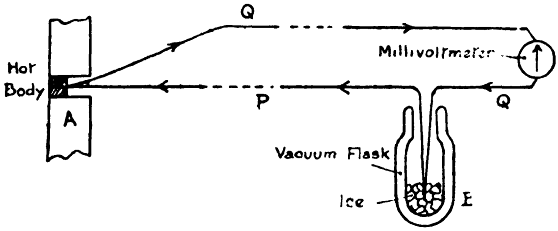


FIG. 300.

the mean temperature is raised still higher (with about the same temperature difference as before), a current in the reverse direction occurs.

This effect is exhibited by most pairs of pure metals, and is termed *thermo-electric inversion*. In the case of metals and their alloys, this effect does not usually occur in the same temperature ranges, so that the latter are frequently used for very high and very low temperatures.

The following table gives particulars of the temperature ranges and approximate thermo-E.M.Fs. of some typical pairs of metals used in practice. In each case the cold junction is at 0° C.

In the case of the rare metal wires used in The Cambridge Instrument Co. thermo-couples, the increase in E.M.F. for 100° C, rise of temperature at the middle of the range, i.e. about 700° C., is, approximately, 1·2 millivolts. For Titan base metal thermo-couples, the increase in E.M.F. for 100° C. rise at the middle of the range (550° C.) is about 4 millivolts, while the corresponding change for iron-constantan is about 5·7 millivolts.

TABLE XV

Thermo-Electric Elements

Element (1)	Element (2)	Hot Junction Temperature	Approximate Thermo-E.M.F.
Platinum	*Alloy.* Platinum, 90 % Iridium, 10 %	1200° C.	7·4
Platinum	*Alloy.* Platinum, 90 % Rhodium, 10 %	1400° C.	4·4
Copper	*Alloy—Constantan.* Copper, 60 % Nickel, 40 %	500° C.	27·8
Silver	Constantan	700° C.	27·6
Iron	Constantan	900° C.	26·7
Nickel	*Alloy—Hoskins.* Nickel, 90 % Chromium, 10 %	1100° C.	10·0

Thermo-couples are available, commercially, in standard ranges of from 0°-300° C. up to 0°-1400° C.

Professor Callendar employed thermo-couples consisting of iron wire soldered at the base of a small hole drilled into the walls of a cast-iron cylinder, and insulated from the sides of the hole by means of mica, in connection with his steam-engine temperature investigations. By utilizing a series of such holes, bored at different depths in the cylinder walls, he was able to ascertain not only the mean but the cyclical temperatures existing at different parts of the cylinder wall.

Thermo-couples are very useful for measuring small differences of temperature at two neighbouring points, and also for high temperatures (platinum and platinum-rhodium thermo-elements).

Owing to their small dimensions and low thermal capacities, thermo-couples are very convenient for "*point*" measurements in heat engines. For rapidly varying temperatures such as these, an automatic contact maker is necessary in order, always, to make contact at the same part of the cycle, for each test. In this manner the temperatures corresponding to each part of the cycle can be plotted out on a crank-angle or time base.

It is essential that the thermo-couple should form part of the solid wall, so that the heat waves of conduction may travel in precisely the same manner as through the solid metal. It is therefore necessary to choose the thermo-couple element, so that the flow of heat through it is the same as through the walls at the same depths.

Thermo-electric pyrometers are supplied, in protective sheaths or bulbs, of steel, quartz, silica, or porcelain, according to the temperature range, and can be used either with needle-indicating

instruments or continuous temperature recorders. Several thermo-couples can be arranged to indicate on the same dial, by means of a suitable switch.

Coker and Scoble Method.—Messrs. Coker and Scoble employed thermo-couples of platinum, rhodium, and platinum-iridium for the explosion temperatures in a 12 h.p. gas engine running at 240 r.p.m.; thermo-couples consisting of cast-iron plugs and wrought-iron wires were used to measure the temperatures at or near the surfaces of the cylinder. Fig. 301 illustrates a typical arrangement of such a thermo-couple applied to the head of one of the valves. It was possible to measure not only the mean temperatures, but the cyclical variations by this means.

A percussion contact-breaker was employed to measure the instantaneous temperatures, and a complete cycle, consisting of twelve equidistant readings, could be obtained in about two minutes.

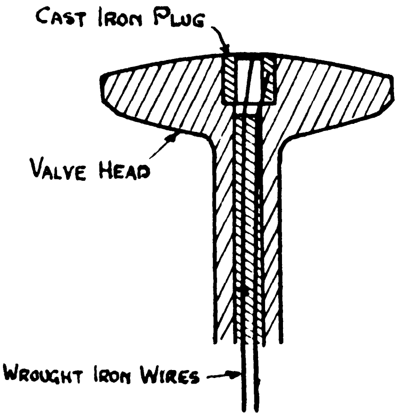


Fig. 301.—Thermo-couple employed for valve head temperature measurements.

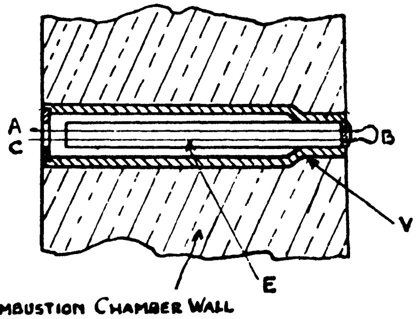


Fig. 302.

The general arrangement of the explosion temperature measuring thermo-couple is illustrated in Fig. 302. It consisted of two platinum alloy wires (of about 0·0005 in. thickness), AB and BC electrically fused together at B, and insulated from the metal body of the valve by fireclay tubes E. Plaster of paris was used in order to keep the whole in position.

The outer steel sheath V, containing the couple, was inserted in the combustion chamber of the cylinder, the conical seating making the necessary gas-tight joint with the cylinder metal. The couple itself in some of the tests was arranged so as to project by about 0·5 inch into the cylinder.

The maximum temperature measured was about 1840° C. during explosion, and at the commencement of the exhaust stroke, 850° C., in the case of an air-gas ratio of 7·35 to 1.

Using a richer mixture of 5·66 to 1, the maximum temperature measured was 2250° C., corresponding to an explosion pressure (obtained from the indicator diagram) of 433 lb. per square inch.

The results of temperature measurements made in this manner for a complete cycle are shown in Fig. 294.

An Improved Thermo-couple.—Although space considerations prevent any detailed accounts of thermo-couple design and construction, mention should be made of an important point in connection with the design of the thermo-couple hot-junction.

Hitherto it was the custom to enclose the two relatively thin wires in an outer sheath in order to protect them from the direct heat and corrosive action of the source of high temperature to be measured, as shown in Fig. 303A. This method has the disadvantage of lag and of a conductivity error resulting in a somewhat lower value of the temperature reading than the correct one.

To overcome these drawbacks the concentric type of thermo-couple shown in Fig. 303B was devised.[1] The time-lag is greatly reduced in this case and accurate readings of the temperature are obtainable.

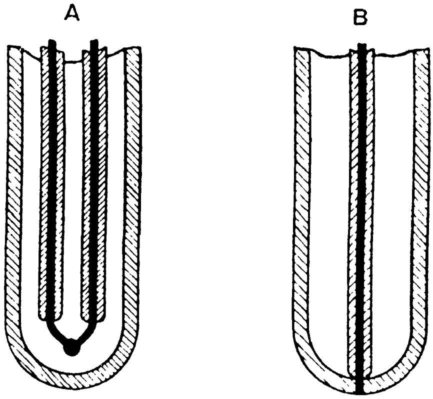


FIG. 303.—Thermo-couple ends.

Method of Measuring Exhaust Valve Temperatures.—The measurement of exhaust valve head temperatures is usually made with thermo-couples inserted into the valve head through the hollow valve stem, in the case of high speed engines. Experience shows that it is not very satisfactory to employ platinum-iridium thermo-couples for this purpose, as they are apt to fracture under the influence of the large inertia forces.

A satisfactory arrangement employed by Dr. Gibson in connection with a lengthy series of valve temperature measurements in aircraft engines [2] is illustrated in Fig. 304. The thermo-couple consisted of steel-Nichrom wires, insulated by means of a refractory cement from each other and from the valve stem.

The temperature measurement in this case necessitated also the use of a " cold junction " and a milli-voltmeter to indicate the potential difference between the hot and cold junctions from which the temperature at the hot junction was deduced. The temperature

[1] " Pyrometry of Exhaust Temperatures of Internal Combustion Engines," C. E. Foster, Diesel Engine Users' Association, May 22, 1925.

[2] See p. 328 (*d*).

of the exhaust valve can also be measured by running the engine with open exhaust and employing an optical type of pyrometer. With the latter it is only necessary to sight the instrument on the hot body to be measured; the light from the hot body enters the

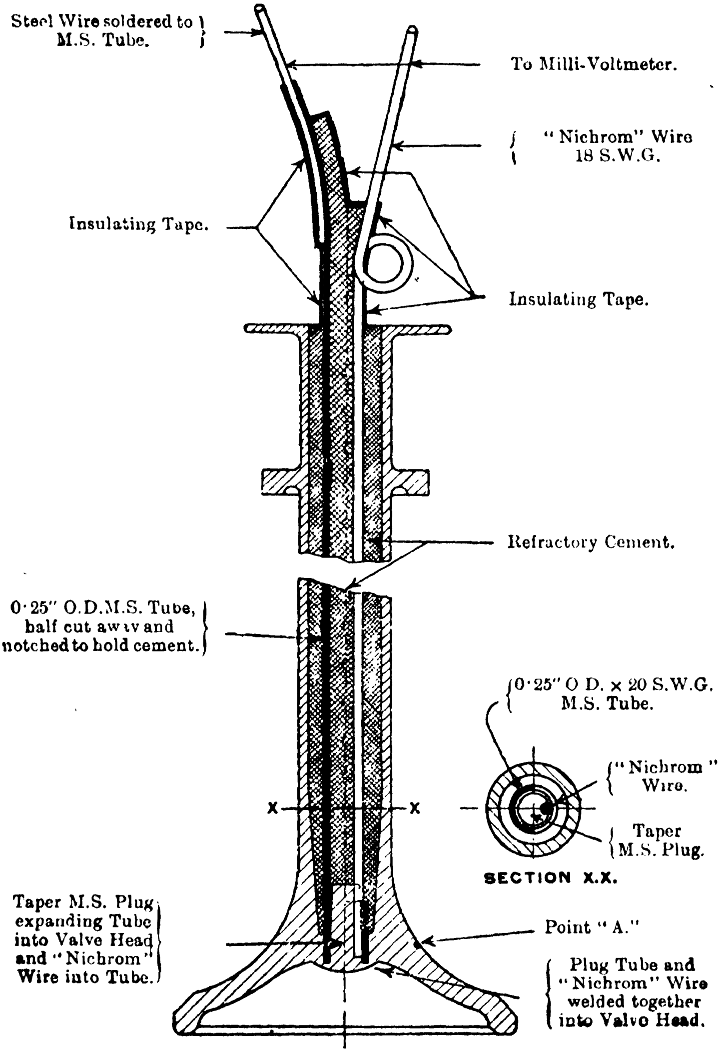


FIG. 304.—Thermo-couple in exhaust valve head.

instrument and passes through an optical system, being finally matched by means of a suitable device with the light from a standard source embodied in the instrument itself. Dr. Gibson employed an optical pyrometer known as the "Pyromike." In these tests the engine was run with open exhaust, and the pyrometer was

sighted through the exhaust port on to the upper side of the (overhead) valve head at a point midway between the stem and the edge of the valve. Under normal conditions this was the hottest part, the periphery of the valve, being cooled by contact with the seating, usually runs " black " in a well-cooled engine.

The results obtained with the Pyromike instrument agreed within about 10° with those obtained by the thermo-couple. The " Shore " Pyroscope was also used at Manchester University by Dr. Gibson. With this instrument the intensity of the light radiated from the heated exhaust valve was matched with the light intensity of a standard paraffin flame modified at will by the use of a calibrated iris diaphragm of red celluloid. The pyroscope was calibrated before use against a standardized thermo-couple inserted in a block of steel in an electric furnace, the readings of the instrument differing only by about 3° C.

The G.E.C. Aircraft Engine Temperature Indicator.—The General Electric Company of America made a very useful type of thermo-couple engine temperature indicator for giving " accurate measurements of the temperatures of cylinder heads or any cylinder hot spots " that may be of particular significance. The instrument is practically instantaneous in operation, and therefore gives immediate indications of the heating or cooling of the engine ; this is an important advantage as compared with an oil temperature indicator which is invariably sluggish in response.

A complete engine temperature indicator consists of a thermo-couple, twin-conductor leads, and a cold-junction, temperature-compensated, remote-indicating instrument. Two different forms of thermo-couples are optional to the purchaser. One consists of a pair of ¼-inch copper-bushed studs to fit drilled holes ; the other is in the shape of a washer, and can be mounted between the spark-plug and cylinder head in the same manner as a gasket, making a very simple installation.

The instrument is compensated for changes in air temperature so that no corrections are required ; incidentally, these corrections are made automatically by a compensator, using a special metal known as G.E. No. 747.

This metal has a peculiar property whereby its magnetic permeability changes with changes in temperature. Placed in the magnetic circuit of the indicating instrument, it is affected by the same air temperatures as the leads and counteracts the errors which would otherwise be introduced.

A selector switch (shown in the centre in Fig. 305) is provided for use where it is required to read successively a number of different thermo-couple temperatures on one instrument. Fig. 305 also shows how these thermo-couples are connected in the case of an air-cooled engine.

Cylinder Temperatures.—The actual temperature at any point on the cylinder walls varies according to the position of the point, and to the part of the cycle at which it is measured.

The temperature, in general, is higher at the combustion chamber end than in the cylinder barrel, and is greater at, or near, the inner surface than the outer one. There is always a temperature gradient across the thickness of the walls, and a small fluctuation of the temperature value at any part due to the variation of heat flow during a cycle of operations.

In the case of a gas-engine cylinder Professor Coker[1] ascertained that the maximum temperature of the walls was only 7° F. in excess of the mean temperature, and at a depth of 0·4 inch from the inside of the cylinder the temperature fluctuation had an amplitude of about $\frac{1}{500}$ of that at the inner surface exposed to the gases. It is a fairly straightforward matter to calculate the temperature amplitudes at any part of the cylinder walls.

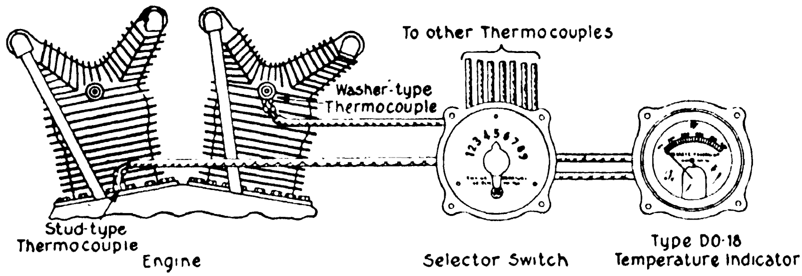


FIG. 305.—Showing arrangement of G.E.C. temperature indicator for air-cooled radial engine.

In the case of high speed engines the surface temperature fluctuation will not be so great as in the case of the slower speed gas engine, due to the higher frequency of the temperature variation and the smaller period available, therefore, for the walls to take up this variation.

It is fairly certain, then, that the amplitudes of the temperature variations at different parts of the cylinder walls will be comparatively small, and of little practical importance. The mean temperature at any point, and the temperature gradient across the conducting metal, such as the walls and fins (in the case of air-cooled engines), are the important items with which the designer and research worker are concerned most.

The mean temperature at any point in the cylinder wall can be measured by means of thermo-couples inserted in small holes, drilled in the walls to the desired depth ; it is necessary to ensure that there is good metallic contact between the head of the thermo-couple and the metal of the walls.

[1] " The Temperatures of the Walls of a Gas Engine Cylinder," *Eng.*, 1908, p. 497.

Reference has been made already[1] to one arrangement for measuring the combustion chamber temperature, in which platinum and platinum-rhodium wires were employed for the thermo-couple.

Dr. Gibson has determined the cylinder head temperatures in the case of an automobile engine by means of thermo-couples of copper constantan, similar to those illustrated in Fig. 306. These were enclosed in screwed plugs, which were inserted in the metal walls from inside the cylinder, after which the heads were ground down flush with the walls. Good, metallic contact was ensured by tinning the head of the couple with solder having a high melting-point. The wires, insulated by enamel or double silk winding, were led through the water-space in small bore rubber tubes, and where these passed through the jacket walls the water-joint was made by a rubber-packed gland. The common cold junction of the thermo-couple was made in an ice-flask. These couples were also employed for ascertaining the temperatures of the water-space and the exhaust valve guide.

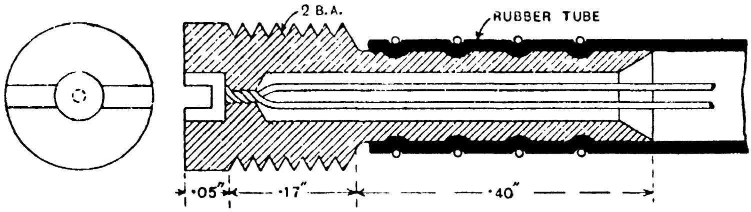


FIG. 306.—Illustrating Dr. Gibson's method of measuring cylinder wall temperatures.

The wires employed for the hot junction of the thermo-couple (Fig. 306) consisted of 30 S.W.G. copper and constantan insulated, and brazed to the mild steel body of the plug. Messrs. The Franklin Manufacturing Company, of America, manufacturers of air-cooled engines, measured the temperatures[2] down the side of one cylinder in nine different positions by inserting thermo-couples inserted in the walls to within $\frac{1}{16}$ inch from the inner surface of the cylinder wall. The temperatures at these places were indicated by means of a Leeds and Northrup potentiometer arrangement, and measurements were also made of the air quantity and velocity and the fuel used during the tests.

Fig. 307 illustrates the positions of the thermo-couples, and shows also the temperatures measured at these places, at different engine speeds (and outputs).

The same method was used also to measure the temperatures on the cylinder head at equal distances of $\frac{1}{4}$ inch from the outside of the wall, and $\frac{7}{16}$ inch below the top of the cylinder head. The results

[1] *Vide* p. 340.

[2] *Automotive Industries*, "Research in Air Cooling," June 8, 1922.

showed that the minimum temperature of about 400° F. occurred almost diametrically opposite to the exhaust port, and the maximum temperature of about 700° F. on the cylinder head nearest the exhaust port, the engine speed being 1800 r.p.m., and output 25·4 h.p.

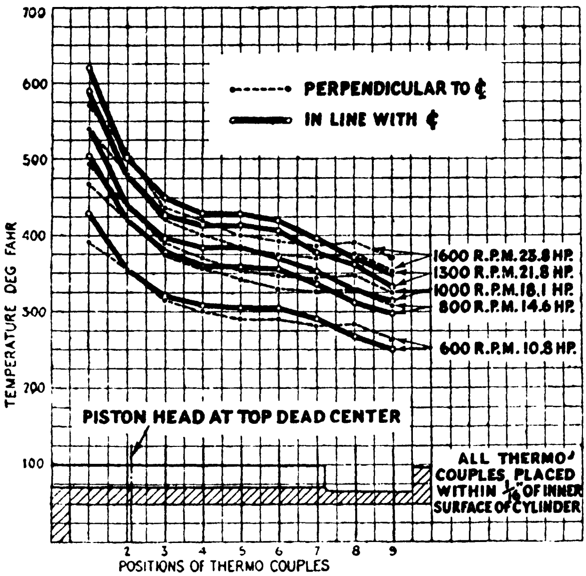


FIG. 307.—Illustrating results of temperature tests on Franklin engine.

A thermo-couple hot junction employed [1] for cylinder head temperature measurements was identical with that shown in Fig. 306. The plugs containing the thermo-couples were screwed tightly

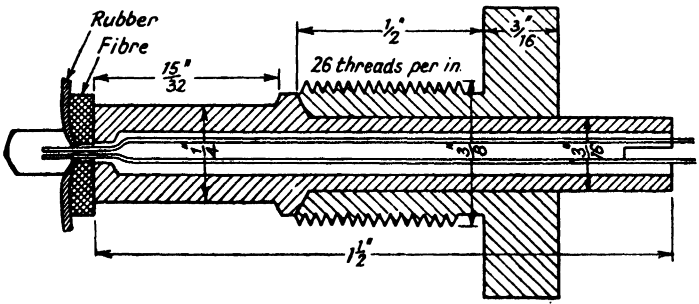


FIG. 308.—Thermo-couple screw and sleeve, showing hot junction and leads.

into the combustion chamber head; the head of the screw was then ground off leaving the remainder flush with the cylinder wall.

[1] " Analysis of Tests on a High Speed Petrol Engine," C. Lyon, *Proc. Inst. Mech. Engrs.*, 1925.

The wires were of copper and constantan of 30 S.W.G. brazed to the mild steel body of the screw. The plug was fitted close to the exhaust valve seating. The two wires from the junction were cut off 4 inches away from the cylinder head, and long 20 S.W.G. insulated copper leads soldered on. Small diameter tubes of rubber were slipped over each of these wires to prevent water from making contact with them and thus causing a short circuit; these were taken out of the cylinder jacket through separate glands.

For measuring the cylinder barrel temperature the screw and sleeve arrangement shown in Fig. 308 was employed.

The wall of the cylinder jacket was drilled and tapped for the reception of the screws at three points in a vertical plane through the crankshaft, two—P and S (Fig. 309)—being on opposite sides of the cylinder at about the level of the highest point reached by the top of the piston, and a third—Q—on the exhaust side at a distance below P of about half the stroke. Holes $\frac{3}{32}$ inch in diameter and $\frac{1}{8}$ inch deep were drilled in the cylinder wall, with the thermo-couple screws as jigs. A somewhat similar arrangement was made to fit a junction at station R just below the bottom of the jacket and vertically below P and Q, as shown on Fig. 309.

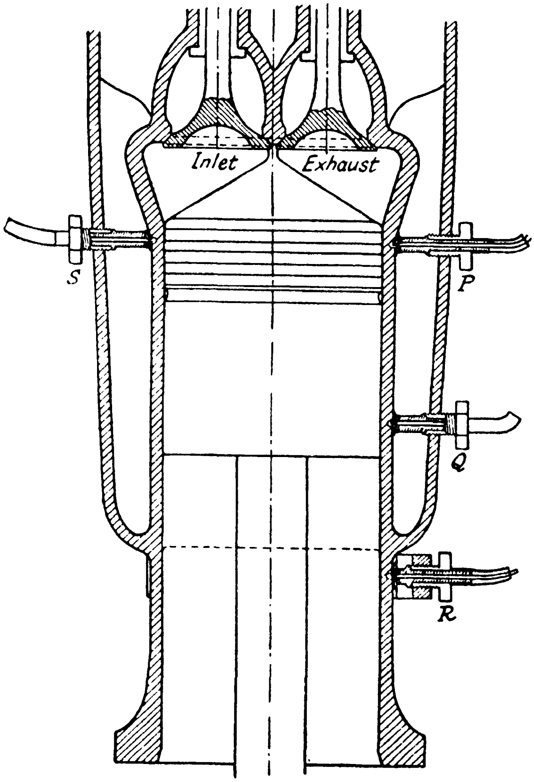


FIG. 309.—Showing location of thermo-couples.

The junctions were formed by melting a blob of brass on the ends of the two wires, 30 g. copper and 24 g. constantan respectively, which were clipped together. This brass end was ground down to a good fit in the blind holes drilled in the cylinder wall.

Thin rubber and vulcanised fibre washers were slipped on over the free ends of the wires, the former to prevent leakage of water to the wires, and the latter to provide thermal insulation between

the cylinder wall and the sleeve. The sleeve was next pushed on, and shellac melted in while the wires were held apart.

Each junction thus made was tested for short circuit, then all four were connected up to a cold junction formed of a copper rod immersed in an ice-flask, and to a switch and milli-voltmeter. Calibration was carried out in steam and in melted tin, and checked at atmospheric temperature.

At the conclusion of the tests the thermo-couples were removed and recalibrated, but no difference from the original calibration was observed.

After calibration the hot junctions were fitted to the cylinder, the sleeves being prevented from turning with the screw by insertion of a screw-driver in the slot at the end. A liberal coating of red lead was spread on as a final precaution against leakage of water. Outside the cylinder the wires were covered with thick rubber tubing to prevent accidental damage during erection of the cylinder.

It was stated that this design of hot junction attachment proved entirely successful, no trouble of any kind being experienced throughout the trials. As regards the accuracy of temperature measurements made by this means, readings of the milli-voltmeter could be taken to ·01 m.v., corresponding to less than ¼° C., and during the trials repeated observations were always within ·03 m.v. of the first, so that results could be assumed accurate to 1° C. The results of cylinder barrel and head temperature tests made on the single cylinder engine of 3½-inch bore and 5¼-inch stroke, with a compression ratio of 5·07 : 1 at 1800 r.p.m., using benzole and Pratts No. 1 petrol, are shown in Fig. 310 ; the B.M.E.P. and thermal efficiencies for the range of mixtures denoted by the abscissæ are also given.

General Requirements for Fixing Hot Junctions.—As a result of experience obtained in connection with the use of thermo-couples for measuring cylinder temperatures, the following conditions have been laid down in regard to the correct method of fixing the hot junctions :—

(1) Good thermal contact with cylinder wall.

(2) Thermal insulation of junction from other masses of metal presenting appreciable surface for radiation.

(3) Electrical insulation of wires from every part and from one another except just at the junction. Water must not be allowed in contact with either wires or junction.

(4) Be such as not to allow jacket water under pressure and at 100° C. to leak.

(5) Must not interfere with the flow of jacket water, or the heat flow at any point in the cylinder wall.

(6) Must not impose any strain on the wires either during fitting or during running of the engine.

(7) Be strong enough to withstand rough usage of the cylinder during erection and alteration of the compression ratio.

Sparking Plug Thermo-couples.—In cases where it is inconvenient or impossible to provide special locations for cylinder temperature measuring thermo-couples, use can generally be made of the sparking plug or its sealing washer. This adaptation of the sparking plug to a dual purpose is of special interest in the case of

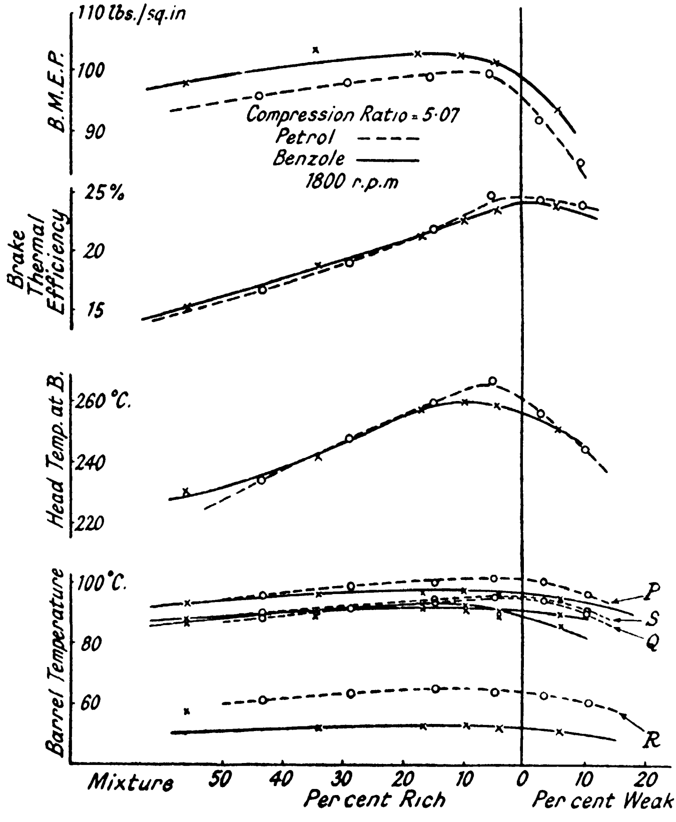


FIG. 310.—Results of petrol engine temperature tests (Lyon).

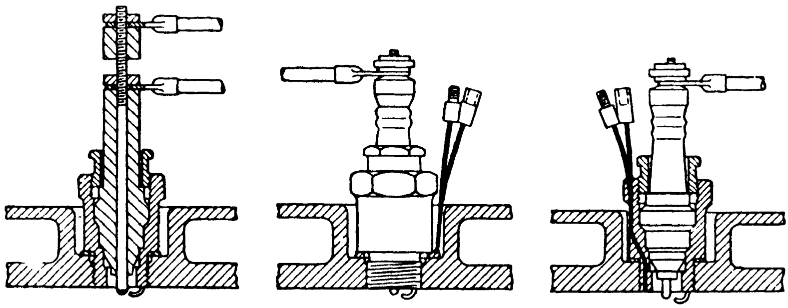
water-cooled engines, and in this connection the three alternative arrangements have been described by G. D. Boerlage and A. G. Cattaneo[1] of the Dutch Shell Engine Research Station, Delft, Holland. The method shown at A (Fig. 311) uses an axially perforated central electrode as one pole of the thermo-couple, the second pole being an insulated wire led through the hole ; this design, however, shows the mean gas temperature around the electrode of the plug.

[1] *The Autom. Engr.*, Feb. 1937.

As in most instances measurement of the temperature of the combustion chamber wall is of more importance, the arrangements shown at B and C are preferable to that of A.

The method illustrated at B, which is *in wide use in aircraft engine test work,* employs a thermo-couple soldered to the sparking-plug washer. This is relatively simple and cheap to instal but it has one disadvantage, namely, that the position of the washer, being nearer to the cooling medium than to the combustion chamber wall, the thermo-couple will respond mainly to fluctuations of the temperature of the cooling medium, so that variation of the cylinder head wall temperature will be recorded on a distorted and reduced scale only.

The design shown at C (Fig. 311) was evolved with the object of avoiding these disadvantages. In this arrangement the insulated thermo-couple wires were introduced through two holes in the body



A B C

FIG. 311.—Sparking-plug thermo-couples.

of the plug and brought into contact with each other in the rim pointing towards the combustion chamber and flush with the surface.

Tests made upon a water-cooled variable compression engine with each of the three designs shown in Fig. 311 showed conclusively that the method C was the most accurate for cylinder wall temperatures. Method A gave very much higher temperatures, of the order 650° to 800° C., whilst method B gave values about 30 to 40 per cent. too low.

Mercury-in-Glass Thermometers.—For engine test work these are often employed for measuring the temperatures of the oil, fuel, air, and cooling water. The temperature range of these thermometers is usually from 0° C. to 100° C. (or 32° F. to 212° F.), but instruments with smaller ranges, giving more open scales, are obtainable commercially. For example, in the Boys Calorimeter used for measuring the calorific values of fuels, the Beckman thermometer has a range of only 5° C., the divisions giving readings of $\frac{1}{100}$° C. In the case of atmospheric temperatures, or liquid fuel

temperatures, a special limited range thermometer of 0° C. to 40° C. will be found useful.

It is necessary to check the accuracy of the readings of the thermometer at intervals. A thorough investigation involves, usually, corrections for calibration errors, zero errors, fundamental interval, pressure, stem exposure, and scale.

For most practical purposes, it is usually sufficient to check the zero and boiling-points by immersing the thermometer in melting ice, and in steam from boiling water. The latter is the more difficult determination, and involves the use of a special jacketed arrangement to ensure that the steam circulates freely around the bulb and stem at atmospheric pressure; a correction for barometric height is necessary.

Some mercury-in-glass thermometers suffer from the defect of irregular capillary tube (or stem) bore, so that the divisions and subdivisions, if equally spaced, or obtained by dividing the stem between the boiling and zero-points into, say, 100 equal divisions, do not indicate the correct temperatures. Comparison with a standard thermometer, or calibration by one of the leading testing institutions, such as the National Physical Laboratory, is desirable where accurate temperature indications are necessary.

Vapour Pressure Thermometers.—Reference has been made already to the applicability of this type, which is generally known as the "transmitting type." It has been mentioned that the bulb is filled with a volatile liquid, such as ether, for temperature ranges of 50° to 100°, say, and with mercury for higher temperatures up to about 500° C.

In the former type it is the vapour pressure of the volatile liquid which is utilized to indicate the temperature. The vapour pressure of the liquid which is volatilized at any temperature increases with the temperature, and in this instrument is arranged to actuate a Bourdon type pressure gauge, which, however, has a scale graduated in terms of corresponding temperatures.

The vapour pressure thermometer is suitable for measuring the temperature of the cooling liquid in both water- and liquid-cooled engines (such as the ethylene-glycol-cooled ones). They are usually made in two ranges, namely, 0° to 100° C. and 30° to 200° C. The former range is suitable for temperature measurements of water-cooled engines and engine lubricating oil; the latter range is applicable to ethylene-glycol liquid cooled engines.

Fig. 312 shows a typical vapour pressure thermometer. It comprises a closed system consisting of an elongated bulb, a capillary tube and a pressure gauge. The system is partially filled with a liquid giving a suitable vapour pressure when heated to the maximum desired temperature. The capillary tubing connects the bulb and the pressure gauge; the tubing can be made several feet in length

for distant reading purposes. The end of the capillary tube extends into the liquid in the bulb so that the pressure gauge Bourdon tube is filled with liquid and not condensing vapour. It is essential in this class of instrument that the free surface of the liquid shall always be within the bulb. Since the vapour pressures of the liquids do not vary uniformly with the temperature, it is necessary, in order to obtain an open scale of temperatures on the Bourdon gauge dial, to employ a suitable type of mechanism for this purpose.

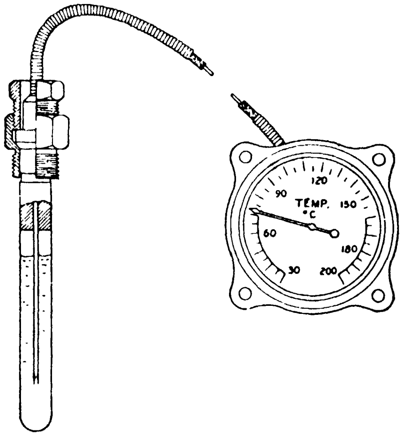


FIG. 312.—Vapour pressure type thermometer.

Among the liquids used for vapour pressure thermometers are sulphur dioxide (0° to 100° C.); methyl ether (0° to 100° C.); methyl chloride °(0 to 100° C.); and ether (0° to 200° C.).

In connection with the *installation of vapour pressure thermometers* it is important to arrange that the capillary tube is not subjected to engine or structure vibration effects or to any chafing or straining action where it joins the bulb. If the capillary tube is left with a free coil or two and is taped near the ends this will reduce greatly the risk of fracture. Local overheating of the tube should also be avoided since large errors in the temperature indications would otherwise be introduced.

Distant Reading Type Thermometer.—Fig. 313 illustrates the Negretti and Zambra expansion type transmitting thermometer. In this it is the liquid expansion which actuates the indicating

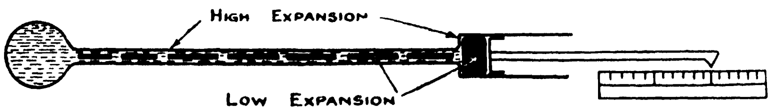


FIG. 313.—Illustrating the principle of the Negretti and Zambra transmitting thermometer.

mechanism (which is usually a Bourdon tube). The bulb, shown inserted in a pipe, is of small capacity, and is connected by means of a capillary tube of high expansion material, with a tempered steel Bourdon tube, which actuates the indicating needle through a worm and quadrant gear. The whole system is filled with mercury, and is, of course, enclosed. The capillary tubing has a wire of low expansion material running throughout its whole length, the dia-

meter of the wire being something less than the bore of the tube. The wire is in convenient lengths (as shown diagrammatically in Fig. 313), so that it can slide in the tube when the temperature changes. The space not occupied by the wire contains the fluid. The volume of the wire and the fluid and the capacity of the tube are so calculated that for a given change in temperature the change in volume of the fluid is the same as that of the space it can occupy, so that no fluid is expelled into the indicator to move the index. A proper compensation can be obtained by this method, and errors due to changes of temperature of the capillary tube are thus avoided.

The dial will give accurate readings at a distance of anything up to 20 or 30 feet from the bulb; it has been used up to 75 feet away with satisfactory results. A test conducted at the National Physical Laboratory on one of these instruments showed that heating 10 feet of capillary 20° C. only affected the indicator by $\frac{1}{10}$° C. The thermometer is also quite sensitive; the N.P.L. tests showed that it took up a change of temperature of 20° C. in 20 seconds.

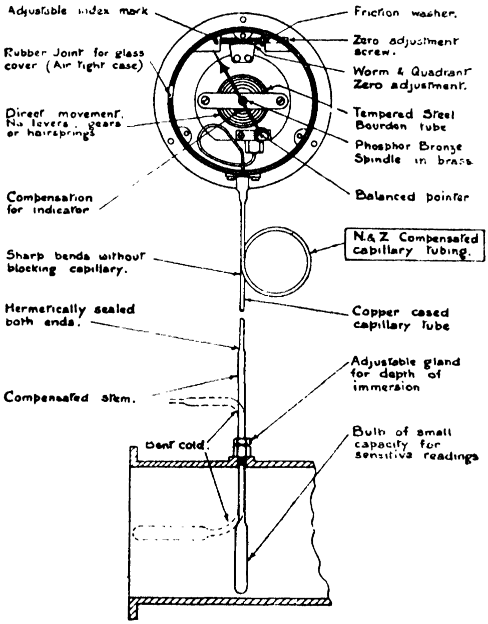


FIG. 314.—The Negretti and Zambra transmitting thermometer.

The accuracy of the instrument was such that under ordinary conditions the reading is accurate to within 1° C., but will usually be found not to exceed $\frac{1}{4}$° C.

The absence of levers and hair-springs is an important advantageous feature of this instrument.

The Exhaust Gas Temperature.—It is sufficiently accurate for most purposes to measure the temperature of the exhaust gases by means of a thermo-couple or resistance thermometer in the near vicinity of the exhaust valve, in order to deduce the temperature at the opening of the exhaust valve. In a high-speed engine of the multi-cylinder type the exhaust frequency is sufficiently high, as a rule, to justify this procedure.

Fig. 315 illustrates an arrangement employed by the author for measuring the exhaust gas temperatures in the case of a

four-cylinder automobile engine. The exhaust manifold was made air-tight, and was lagged with asbestos to prevent any appreciable radiation of heat. A platinum resistance thermometer was inserted into the exhaust manifold so that the resistance element was very close to the exhaust valve of No. 1 cylinder. The stem of the thermometer was made gas-tight by means of an asbestos-packed gland.

The platinum resistance thermometer was first calibrated on a Callendar bridge, and then on a Harris recorder which was used

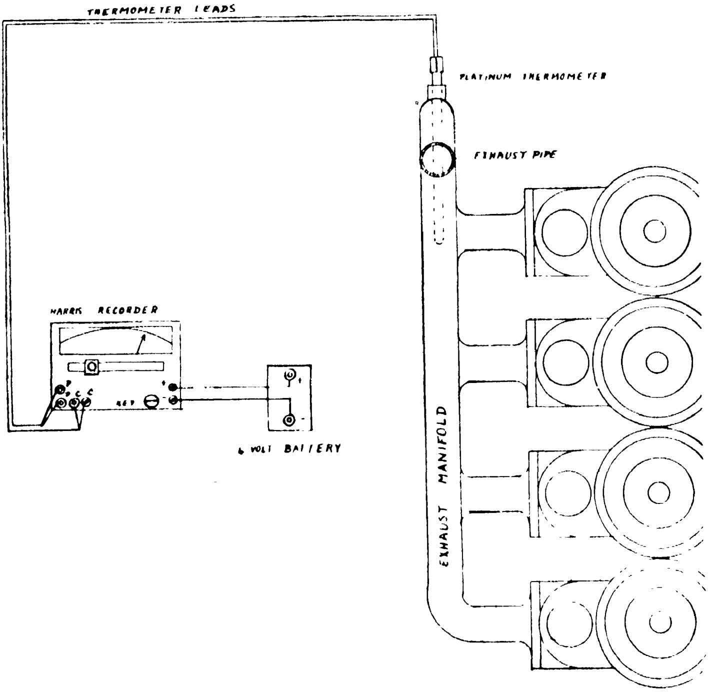


FIG. 315.—Showing method of measuring exhaust gas temperature.

thereafter. The " fixed points " of the thermometer were determined in melting ice, steam, and at the boiling-point of sulphur.

It was necessary, before taking readings, to allow the engine to warm up to its normal working temperature. The results of a series of mixture variation tests obtained in this manner are shown in Fig. 293.

The exhaust gas temperature can be computed, also, from the indicator diagram, but a knowledge of the suction temperature is necessary. Professor Watson [1] employed this method to ascertain the exhaust temperature in connection with certain fuel researches.

[1] " Benzole, Alcohol, and Mixtures with Petrol as Fuels for Internal Combustion Engines," W. Watson, *Proc. Inst. Autom. Engrs.*, 1914-15.

He assumed a suction temperature of 147° C., and then calculated the exhaust temperature corresponding. Next, from the measured amount of fresh charge at the temperature at which it was taken in, he deduced the suction temperature, allowing for the admixture of this fresh charge with the residual exhaust gases at the calculated temperature. Finally, with this "corrected" suction temperature, in conjunction with the indicator diagram he was able to calculate the temperature at any point on the indicator diagram. A correction was necessary for the volume increase of the gases after combustion.

The suction temperatures obtained in this manner varied from 125° C. to 145° C.

The Cambridge Diesel Engine Pyrometer.—In most experimental and in many stationary Diesel engine installations exhaust gas pyrometers are provided for the purpose of affording a reliable indication of the combustion conditions and also to show when any incipient trouble occurs, such as jacket failure, injection variations or stoppages, etc.

The Cambridge exhaust pyrometer employs the thermo-electric method of temperature measurement, the current being measured on a galvanometer provided with a scale calibrated in temperature

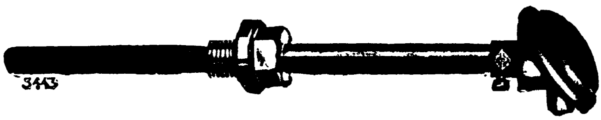


FIG. 316.—Cambridge exhaust pyrometer.

degrees. The instrument possesses the advantage that all the thermocouples in an installation can be connected to one central indicator placed at the most convenient position, so that the temperature of the gases in each exhaust pipe can be accurately read off in turn from the one instrument by simply rotating a switch. The advantage of this centralized method is obvious. There is no need to approach each thermometer separately to obtain accurate readings, and the temptation to depend at times upon a cursory glance along a number of dials more or less widely separated is avoided. By turning the switch to each position in succession, the behaviour of each cylinder can be closely watched.

The indicator may be arranged either for wall mounting or for flush mounting on a panel. The indicator itself is mounted in the upper part of the case, and has no external projections ; it has a scale 5 inches long, with open graduations and large figures which, together with the bold pointer, facilitate correct reading.

The standard ranges are 0° to 1200° F. and 0° to 600° C. A multi-way selector switch, which may connect either 10 or 20 points, is operated by an external handle, and is housed in the lower compartment, into which pass the leads from the thermo-couples.

The thermo-couple consists of the two dissimilar wires, protected by a steel tube with collar screwed $\frac{3}{4}$ inch gas. The wires are brought along the tube to an enclosed head which accommodates terminals for connection by compensating leads to the indicator. The thermo-couple is screwed into the exhaust pipe at the required position ; the length of stem (*a*) between head and screwed fitting, and also the length (*b*) below the stem, projecting into the exhaust pipe, can be varied to suit requirements.

Temperature Measurements in Altitude Laboratory Tests.—In Chapter XII a short account is given of the American altitude laboratory for testing engines under different conditions of air density, pressure, or temperature. In connection with these tests, a very complete equipment for temperature measurements of the different factors was provided. Twelve thermo-couples in all were employed.

The leads from the thermo-couples passed through an opening in the side wall of the altitude chamber to a table on which were mounted the necessary switches and potentiometer. The galvanometer was swung in a special cradle mounted on a solid concrete pier to eliminate so far as possible the effects of vibration. Considerable difficulty was experienced in the early operation of the plant owing to the lack of a proper support for the galvanometer. The vibrations from the engine were transmitted through the ground, so that even the concrete pier was not sufficiently steady, but the arrangement adopted later did away with this trouble to a large extent.

The thermo-couples were all copper-constantan couples, the junctions being made by twisting the ends of the wires together and soldering with silver solder. The set of couples was made up as follows : An ice junction common to three junctions placed in the oil pipes ; an ice junction common to seven junctions used about the carburettor and as spares ; an ice junction and a junction suspended in the chamber to measure the room temperature ; an ice junction and one junction inserted in the carburettor air line above the Venturi ; an ice junction and two junctions placed in the outlet and inlet jacket water pipes ; and outside the chamber a circuit of three junctions ; one in the water main and two in the exhaust tanks.

All couple wires were wrapped with tape for insulation to prevent rubbing and to give them " body." They were connected with rosin-soldered joints to copper leads passing through the wall of the chamber and brought out to a dial switch with all copper contacts. Before being fastened to the switch, each couple had a small coil of manganin wire connected in series and adjusted to give it an equal resistance to all other couples. The switch had a rotating arm which carried two copper spring contacts which bore in turn upon the copper contacts to which the couples are attached.

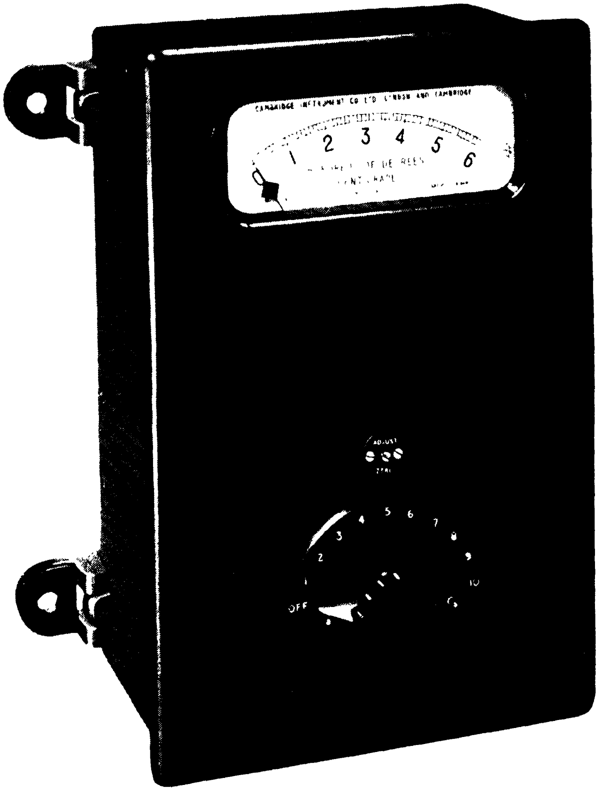


FIG. 317.—The Cambridge exhaust temperature indicator

[To face page 356.

From the arm the circuit was completed through a carefully made manganin resistance of 0·029 ohm, a manganin dial resistance box adjustable in steps of 0·1 to 1000 ohms, a galvanometer and a key. The 0·029 ohm resistance formed part of a potentiometer circuit consisting of a dry cell, a milliammeter, a double-throw switch, and a slide wire rheostat. Another branch circuit was controlled by the closing of a double-pole switch, which served to connect a carefully balanced pair of resistance coils of approximately 85 ohms each, and these, together with the dry cell, galvanometer resistance box, and thermal element, formed a Wheatstone bridge.

The potentiometer afforded a means of measuring the sensitivity of the galvanometer as follows: Across two terminals of the dial switch a coil was connected having the same resistance as the thermal elements but without their thermo-electric property. The arm was placed on these terminals, and the sensitivity was then found by establishing a definite current through the potentiometer (50 milli-amps) and observing the galvanometer deflection. This was found subject to variation on account of the heavy machinery in its neighbourhood, but could always be brought back to its original value by an adjustment of the resistance box.

The Wheatstone bridge served to compare the resistance of the different couples up to the point of attachment to the dial switch and was sensitive to 0·1 ohm.

Thus the E.M.F. of any thermo-couple could be measured by the current it established through the galvanometer; and with the bridge and potentiometer the conditions of measurement could be made equal for all couples and held constant at all times.

Electric-resistance Thermometers.—Although the thermo-couple can be employed for measuring moderate and high temperatures, there is an alternative means in the platinum-resistance thermometer. The latter is exceedingly accurate, and can be employed over a wider range of temperatures. It is possible to measure temperatures up to 500° C. with an accuracy of $\frac{1}{100}$° C., and up to 1300° C. with no greater error than $\frac{1}{10}$° C. with this type of thermometer. It occupies a rather greater bulk than the ordinary mercurial or thermo-couple types, so that it is not convenient, in general, for use in high-speed engine testing, where temperatures at a point, or where local temperatures, are concerned.

It is particularly suitable, however, for temperatures beyond the range of the mercurial type, such as those of exhaust gases. The principle of this type depends upon the increase in the resistance of pure platinum wire with increase of temperature. The resistance-temperature relation is given by

$$R_t = R_o(1 + at + bt^2)$$

where R_t is the resistance at $t°$ C. and R_0 that at 0° C.; a and b are constants.

The values of R_0, a and b can be determined by measuring the resistance (R_t) at three known temperatures (t). The most convenient temperatures are the freezing-point of water, its boiling-point, and the boiling-point of sulphur (444·53° C. at standard pressure).

Professor Callendar developed the platinum resistance type of thermometer to a high degree of accuracy. This method of using the thermometer is to employ a Wheatstone type of bridge as shown in Fig. 318.

The platinum wire is wound on a mica frame, and is connected by copper leads to terminals on the further end of the thermometer porcelain tube. The platinum wire R and leads form one arm of the bridge. An identical pair of copper leads C (or, rather, a length of wire equal to that of the other leads) is placed in the same porcelain tube, but insulated from R, and inserted in the other arm of the bridge. The object of these leads C is to compensate, automatically, for the variation in resistance of the other leads.

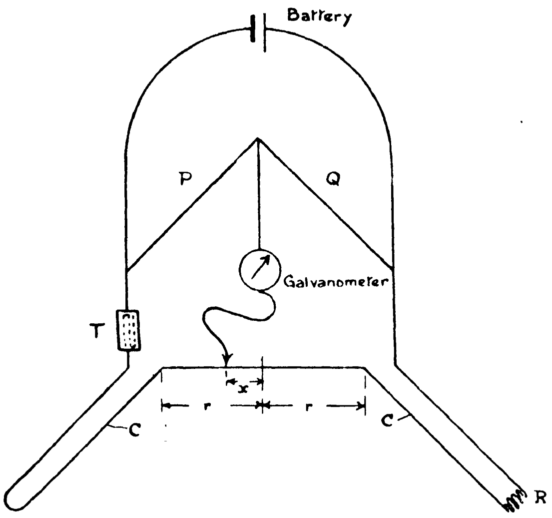


FIG. 318.—The electric resistance thermometer.

In Fig. 318, if R represents the resistance of the platinum wire, T that of the resistance coils, C that of the leads C, and if X is the position of the sliding contact on the potentiometer, for electrical balance, as shown by the sensitive galvanometer G, then since the resistances P and Q of the fixed arms are always arranged so as to be the same (i.e. P = Q), we have

$$R + C + r + x = T + C + r - x$$

or

$$R + x = T - x$$

$$R = T - 2x.$$

From this relation the resistance R is determined. This type of thermometer can be provided with an indicator needle and dial, upon which the temperature is read direct. It can also be used with a continuous drum-type recorder for giving a continuous inked

record of the temperature. Similarly, to the thermo-couple, it possesses the advantage that it can be placed at long distances from the indicator/or recorder, and a number of thermometers in different places can be coupled up to one indicator.

In this manner an operator in a central office can ascertain the temperatures of various items which may be scattered all over the works.

The upper useful limit of the platinum resistance thermometer is about 1200° C., and the lower one $-200°$ C. It is necessary to employ a glazed porcelain sheath for use at the highest temperatures, fused silica up to 1000° C., and copper or steel up to about 800° C. for protective purposes.

The length of the sheath is usually arranged in convenient sizes from 280 to 1650 mm.

Special precautions must be taken to protect the fine wire coil from appreciable vibration, shocks, and corrosive fumes.

Chapter XI

COMPRESSION-IGNITION ENGINE TESTING

Tests upon compression-ignition engines may be grouped into two main classes, namely, those upon production types, and those made in connection with research and development work.

The production engine tests usually comprise brake horse-power, fuel-injection equipment, fuel and oil consumption and exhaust temperature measurements.

Research and development work upon this type of engine is generally on the lines of tests on new designs of combustion chambers, fuel-injection nozzle types and locations, and fuel-injection equipment.

These investigations involve the use of suitable designs of indicators, such as the cathode ray one, in order to obtain indicator diagrams on crank-angle or piston-stroke bases and rate-of-pressure rise diagrams during the injection and early part of the combustion periods. Corresponding diagrams of fuel nozzle valve lift and also fuel line pressures give practically all of the information required by the designer.

Both visual and photographic records of combustion pressures, fuel pressures and nozzle valve lift are usually obtained for this purpose ; some typical examples of diagrams taken on compression-ignition engines are given in Chapter VIII.

Ignition Lag.—An important item in connection with tests made upon fuel-injection systems and combustion chambers, as well as on various grades of fuel is that of the *ignition lag* or *delay period*, that is, the period of time that elapses between the moment of the commencement of fuel injection into the combustion chamber and the ignition of the fuel. This delay period has been found [1] to depend upon several factors, including the charge temperature, engine speed, degree of air turbulence, amount of injection advance, compression ratio, etc. A detailed study of the principal causes of ignition lag has proved of great assistance to designers of compression-ignition engines in indicating the effective measures that can be taken in order to vary this delay period or to control it in such a manner that excessive rates of pressure rise—which are associated with " Diesel knock "—are avoided.

The ignition lag in any particular engine is determined by finding the crank angle at which injection commences and the point on the indicator (or pressure-rise) diagram where the pressure line departs

[1] A full account of experimental investigations is given in " High Speed Diesel Engines," A. W. Judge (Chapman & Hall, Ltd.).

from the compression pressure line. The cathode ray indicator has proved very useful for such investigations.

An alternative method of determining the ignition lag[1] is illustrated in Figs. 319 and 320; it was devised to overcome the drawback that the evidence required is not obtained from indicator diagrams at the moment of the event but only later on.

The method, due to A. L. Bird and S. G. Bauer, allows a continuous observation of the crank angle at which fuel injection begins and that at which ignition of the fuel commences; the interval between these two events is the ignition lag.

The impulse which marks the timing of the injection is taken from the automatic needle valve through a very light and stiff leverage; this makes it possible to close a contact within the first 5 per cent. of the needle lift, which experience shows to be sufficiently accurate. The second impulse is taken from an ionization gap, substantially like a spark plug, inserted in the combustion chamber at a place where the beginning of the explosion can reasonably be expected (see Fig. 319). The authors' reason that there is as good justification for taking the time at which the flame first reaches the ionization gap for the beginning of explosive combustion as for taking the point of an indicator diagram at which the rate of pressure rise reaches a suitably defined value.

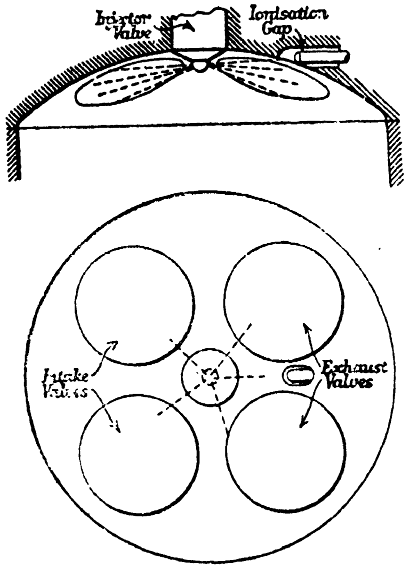


FIG. 319.—Location of the ionization gap in combustion chamber.

The electric current obtainable through an ionization gap is much too small to operate an indicator directly, and for this reason the mercury-discharge valve or thyratron is used for amplification purposes. Its main characteristic is that a very small positive change in grid bias, requiring only an infinitely small current, suffices to release a discharge of the order of a kilowatt, which can be discontinued only by some outside means. Fig. 320 shows the amplifying circuit. The discharge from the thyratron is sent through an ordinary neon lamp l, which is observed through a slot s in the drum d revolving at engine speed. When the flame reaches the ionization gap i the discharge of the thyratron is started and

[1] *Engineering*, Aug. 14, 1936.

the neon lamp lights up. In the stroboscopic drum this is seen as the beginning of a band of light. When at the next working stroke the fuel-injector valve *v* begins to lift, a contact *a* is closed, shorting the thyratron-neon lamp circuit across a condenser *e*. This is sufficient to interrupt the discharge until the next explosion starts it up again. On the stroboscope this appears as the end of a band of light. The length and location of the dark space between the stop and the start of the band of light correspond to the duration and the timing of the ignition lag, measured on a scale *m* divided into degrees of crank angle. A further set of contacts *b* on the camshaft serves to discharge the condenser and to put out the neon lamp during the idle revolution between two working revolutions

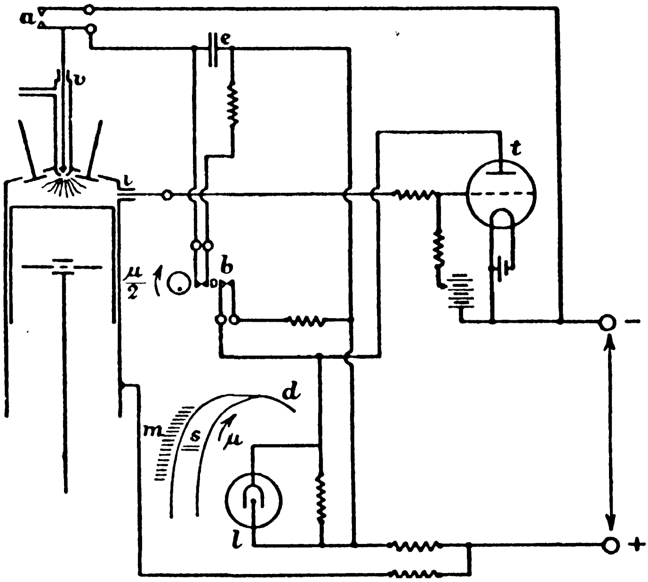


FIG. 320.—Circuit diagram of ignition delay indicating apparatus.

in a four-stroke engine. This is done without interfering with the thyratron, which must be kept burning until the next injection stops it.

Single Cylinder Engine Tests.—For research and development purposes it is usually sufficient to carry out investigations upon a single cylinder engine unit, as in aircraft engine research work. The unit in question can be fitted with different cylinder heads and fuel-injection apparatus for the purpose of combustion phenomena studies, horse-power tests and other measurements.

An excellent example of this type of test equipment is that employed by Messrs. Associated Equipment Company, Ltd., Southall, Middlesex. The experimental engine test-house contains several

rooms for testing complete engines ranging from 70 to 200 B.H.P., and single cylinder engine units. One of the latter test-stands can also be used for the testing of fans, water pumps, exhausters and other auxiliaries.

The multi-cylinder test-beds are equipped with electric dynamometers capable of being operated at speeds up to 4000 r.p.m. In addition there are flowmeters, tachometers, and other measuring devices. The dynamometers can also be used for measuring frictional losses of engines. The single cylinder test-stands employ a B.T.H. electric dynamometer, and in connection with certain tests Standard Sunbury indicators are employed ; in this way records of cylinder pressures, fuel oil pressures in the injection system, and the motion of mechanical parts such as injector nozzle valves can be obtained.

Provision is made for controlling the temperature of the cooling water and also that of the lubricating oil and for the attachment of several indicating elements to the engine. Exhaust gas analysis is carried out with an " Orsat " apparatus. Other equipment includes pyrometers for exhaust gas temperature measurements and thermo-couples which can be arranged to measure in turn cylinder head temperatures at various internal locations, and also to measure bearing temperatures. In connection with *the exhaust gases*, apart from obtaining analyses and temperature measurements, a permanent record of the colour can be made with special apparatus developed by the research staff. The study of exhaust colour is an important item from the point of view of road transport engines, and visual observations are apt to prove misleading. The apparatus employed consists of a simple arrangement whereby a fixed quantity of exhaust gas is drawn through a filter paper so that its carbon particles are deposited on the latter.

Air consumption measurements are made with an Alcock viscous flow air meter, of the type described in Chapter IV, made by Ricardo. This type possesses certain advantages over the orifice, Venturi and similar air meters, especially for use in cases where the air-flow is of a pulsating nature, as with single cylinder engines. The principal advantages are that a large damping capacity is not required between the engine and the meter, and since the manometer readings are directly proportional to the flow, an open scale is obtained, resulting in easier and more accurate readings. The manometer used is of the sloping tube type.

Fuel-injection Test Equipment.—The complete fuel-injection equipment, comprising the fuel-injection and fuel-feed pumps, injection nozzles and filters, may be tested as a whole, using one of the available injection equipment test benches, such as the C.A.V. or Crypton, or individual units can be investigated separately.

The fuel-injection pump is usually tested for fuel quantities

delivered from each plunger unit over a given number of strokes and for fuel-injection advance angle. Fuel-injection nozzles are tested for spraying characteristics, using a stroboscopic device.

The C.A.V. injection equipment test-bench, shown in Fig. 323, enables tests to be made on fuel-injection pumps, pump governors and injection advance devices. The machine, which was described in *The Engineer*, December 30, 1938, is fitted with a wood bench having a gauze-covered drain hole. The heavy base-plate casting carries a gear-box, bearings, a pump-fixing device, tachometer, and counter drives, and a frame support for the testing nozzles. The driving motor B is mounted on a cross-bar underneath the bench, the drive being taken to the gear-box C by means of special belting.

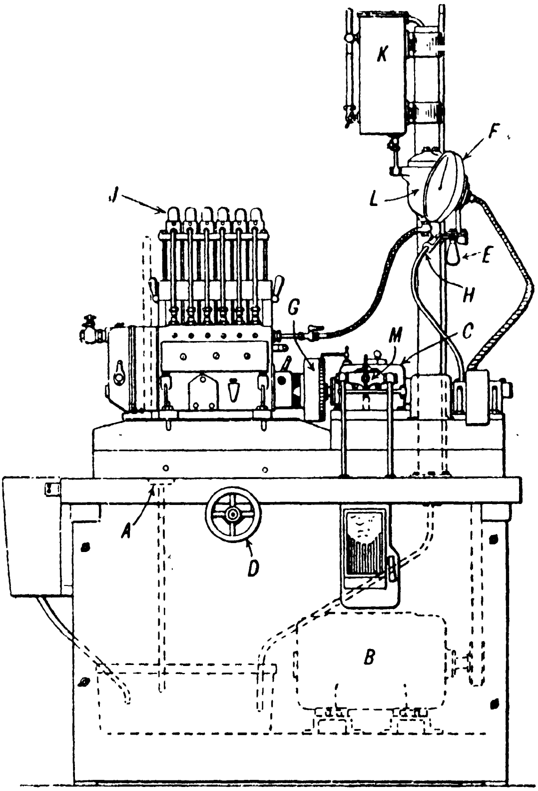


FIG. 323.—The C.A.V. fuel-injection equipment testing apparatus.

When an A.C. motor is fitted, its speed is controlled by means of the hand-wheels D projecting from either side of the bench. Each wheel can be used independently, for they are both mounted on the control rod, which actuates through the medium of a chain and sprocket the movable brush gear of the motor. The motor rotation can be reversed by turning the operating handle in the opposite direction to that for speed increase, until it is past the neutral brush gear-point. In the neutral brush gear position the motor is stationary. Therefore in order to avoid burning out the motor, a red pilot lamp E, fixed to the main support pillar, remains alight as a warning when the motor current is switched on.

Mounted between the driving pulley bearings and the fuel pump is the gear-box, provided with a lever which can disengage the drive,

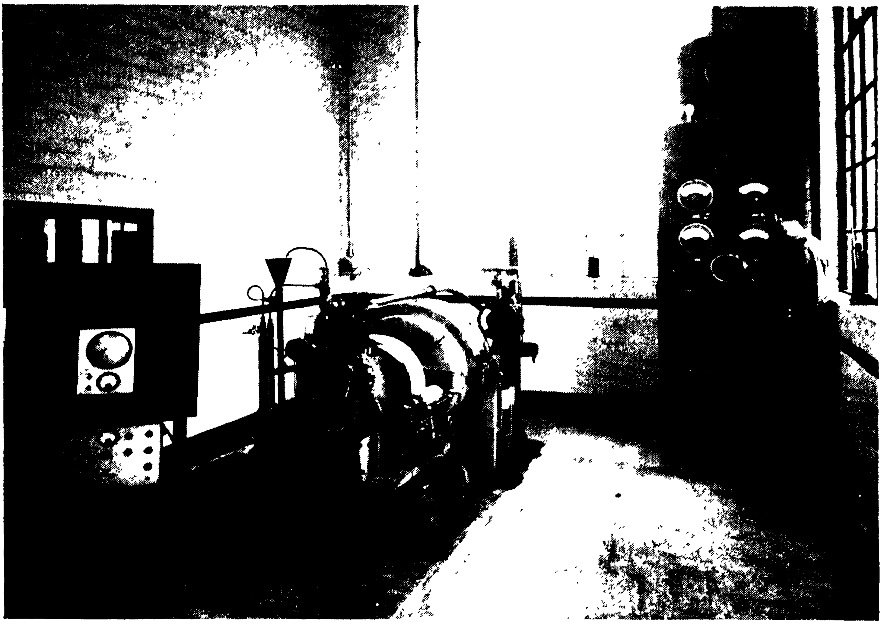


FIG. 321.—The A.E.C. single cylinder C. I. engine test-bed

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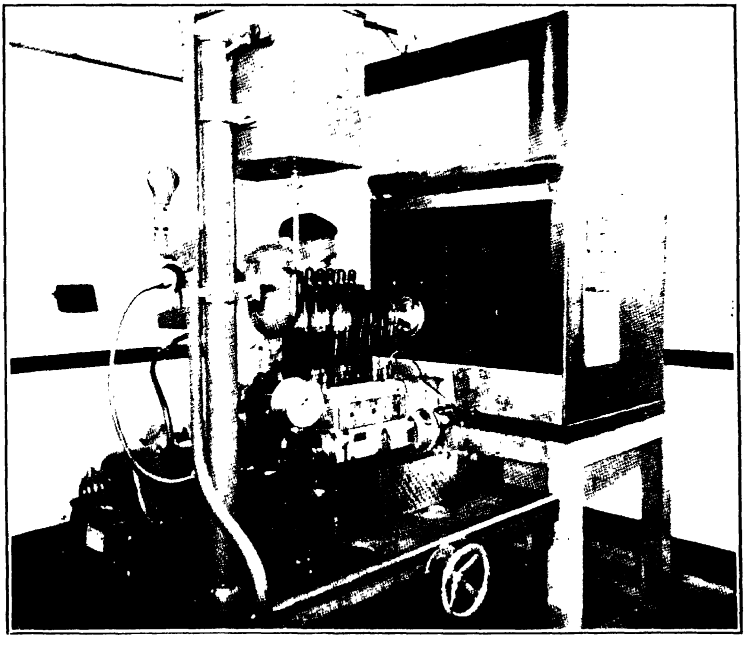

FIG. 322.—Testing a standard six-cylinder engine fuel pump at A.E.C. testing laboratory

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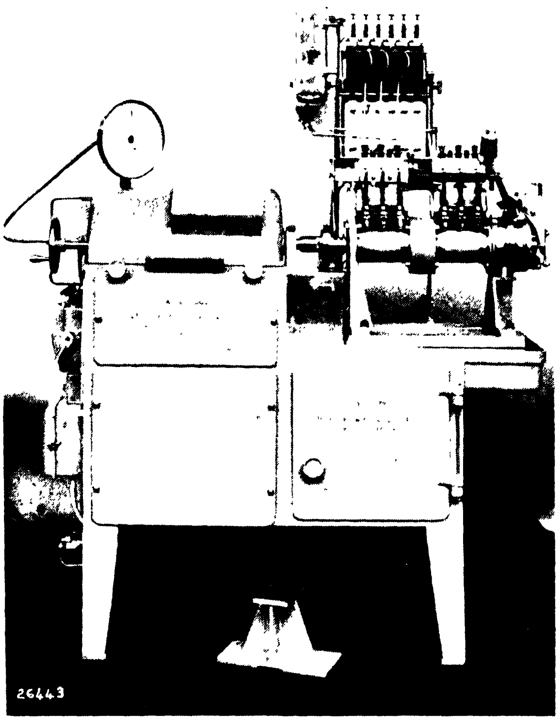

FIG. 324. The Hartridge fuel-pump test-bench

To face page 365.

or provide a direct connection between the pump and the pulley, or reduce the pump speed in the ratio of $3\frac{1}{3} : 1$. A tachometer F is driven by a flexible shaft from gears on the end of the gear-box. It has a speed range of 2000-0-2000 r.p.m., and indicates the actual number of revolutions of the fuel-pump camshaft irrespective of the gear-box ratio. An injection timing device consists of a flywheel G mounted between the gear-box and pump drive. The periphery of the flywheel is graduated into degrees and an adjustable pointer is fitted. Mounted below the tachometer is a revolution counter H, also driven by a flexible shaft from the gear-box.

Testing nozzles J in their holders are mounted on a frame which is clamped to the base-plate casting, and the amount of oil delivered by each nozzle during a predetermined number of injections can be measured by means of graduated glasses mounted below the nozzles. The equipment is completed by a fuel tank K, fitted with an oil level glass, and a standard C.A.V. fuel filter L. Fuel is lifted from the container beneath the bench to the main fuel tank by means of a feed-pump M, mounted on the side of the gear-box and driven therefrom.

The apparatus is so arranged that any single pipe from the fuel-pump can be connected to a fuel injector spraying into a cabinet. The cabinet is fitted with a roller blind to close the front and is furnished with shutters which can be opened for visual examination of the fuel spray, the spray being withdrawn through the pipe at the top of the cabinet. On the far side of the cabinet there is arranged projecting through an opening the neon lamp of a portable stroboscopic device known as the " Strobotac " which is calibrated to read revolutions per minute. It is illuminated by a neon lamp which focuses the light at a distance of about 8 inches from the instrument. The speed of the light flashes, which, in some cases, is as short as three-millionths of a second, is controlled by a knob which also turns an illuminated calibrated dial, reading from 600 to 14,000 flashes per minute.

The Hartridge fuel-pump test-bench [1] is another well-designed piece of apparatus suitable for use in compression-ignition engine reconditioning shops, test-houses, or laboratories. It enables the outputs of individual pump elements to be measured and adjusted to equality; governing mechanisms to be checked, and fuel-injection timing angles to be measured accurately. A complete test upon a six-cylinder fuel-injection pump and accurate resetting takes about an hour with this test-bench.

The test-bench (Fig. 324) comprises a 3 h.p. A.C. or D.C. electric motor running at 1440 r.p.m., provided with an infinitely variable expanding pulley and belt drive giving speeds between 50 and 1500 r.p.m.; it can, however, be set to give a variation from 50 to 4000

[1] Crypton Equipment, Ltd., London, N.W. 10.

r.p.m. if desired. A hand-wheel on the side of the casing controls the speed. Other components include a self-contained electric control gear, pump test platform, tachometer, fuel tanks, filter mechanism, and oil equipment. A special feature is *the automatic trip gear* coupled to the main pump drive, which controls the start and finish of pump output measuring tests which are carried out for exactly 200 revolutions of the pump shaft. For this purpose two cams are provided, one operating only in a clockwise direction, and the other when the machine is required for testing anti-clockwise drive pumps. For output tests two standard sets of test tubes, 8 to 26 c.c., are provided ; one set of tubes is drained whilst the other set is in use.

The *phase angle test gear*, for measuring the exact angle at which injection commences, consists of a special shaft which carries a spring, the shaft being extended so as to project through the top of the fuel injector. Fixed to the shaft is a stirrup to the upper part of which is fitted a tungsten contact. A second contact is fitted on a bronze member on top of the injector, the latter contact being mounted just under the stirrup contact. The lower contact is adjustable so that immediately the injector valve opens, the stirrup follows the valve and immediately breaks the two contacts. This " make-and-break " device operates a high tension coil circuit whereby a spark is produced on a rotary annular spark ring calibrated in 360° and mounted on the test-bench casing. The spark thus shows the exact position when injection commences in relation to the crank position. The test-bench has a selector switch numbered 1 to 6 so that any of the injectors can be tested for injection angle.

In connection with the governor test, readings are obtained from the chisel-pointed level which was fitted to the pump control rod when the unit was first set up for testing. On the nose of the lever is engraved a hair line, which indicates against a metric scale. The scale is universally adjustable, so that it can be set to suit accurately the position of the lever in each individual test.

The test is carried out as follows : The motor is started up and the speed gradually increased until the governor comes into action. A slight movement of the variable speed control will now cause the pump control rod to move backwards and forwards, and, with it, the chisel-pointed lever. Thus, by observing the movement of the hair line against the metric scale, the exact speed at which the governor cuts in or cuts out can be rapidly and accurately ascertained, and the amount of movement directly read off ; also the actual r.p.m. at which the governor operates can be obtained both in coming up to—and in reducing—speed.

Recording Results of Fuel-pump Tests.—A convenient method of recording the results of fuel-pump tests is indicated in

Fig. 325. The particulars required are entered both before and after each test. From the readings on the check sheet records the relationship between the pump test and miles per gallon of fuel obtained on a vehicle can be ascertained, thus giving a useful standard to which pumps can be adjusted to give the best performance.

Individual Injection Nozzle Tests.—In connection with the routine testing of fuel-injection nozzles special testing apparatus now available, such as the C.A.V., Bryce and Armstrong makes, enable the spraying action of any particular design to be studied in the open air, or—with a special auxiliary chamber—under compression ; in addition, the fuel pressure for spraying can be regulated

MAKE OF PUMP	NO	DATE OF LAST TEST	DATE PUT IN SERVICE
		CARD NO.	PERIOD
MILEAGE	MILES PER GAL	DATE OF TEST	TESTED BY
DIRECTION OF ROTATION	TEST OIL VISCOSITY	NATURE OF OIL TEST	TEMPERATURE °F

PUMP OUTPUT READINGS

SPEED R P M	RACK OPENING	No 1 PLUNGER	No 2 PLUNGER	No 3 PLUNGER	No 4 PLUNGER	No 5 PLUNGER	No 6 PLUNGER
BEFORE ADJUSTMENT							
	MM	CC	CC	CC	CC	CC	CC
	MM	CC	CC	CC	CC	CC	CC
AFTER ADJUSTMENT							
	MM	CC	CC	CC	CC	CC	CC
	MM	CC	CC	CC	CC	CC	CC

GOVERNOR TEST — GOVERNOR NO. & TYPE.

BEFORE ADJUSTMENT			
CUTTING IN IDLING SPEED R P M	RACK	CUTTING IN MAX SPEED R P M.	RACK
CUTTING OUT IDLING SPEED R P M	MOVEMENT C M	CUTTING OUT MAX SPEED R P M	MOVEMENT C M
AFTER ADJUSTMENT			
CUTTING IN IDLING SPEED R P M	RACK	CUTTING IN MAX SPEED R.P M	RACK
CUTTING OUT IDLING SPEED R P M	MOVEMENT C M	CUTTING OUT MAX SPEED R P M	MOVEMENT C.M

PHASE ANGLE READINGS

SPEED R P M		No 1 PLUNGER	No 2 PLUNGER	No 3 PLUNGER	No 4 PLUNGER	No 5 PLUNGER	No 6 PLUNGER
BEFORE ADJUSTMENT							
	ERROR	°	°	°	°	°	°
AFTER ADJUSTMENT							
	ERROR	°	°	°	°	°	°

REASON FOR RETURN OF PUMP	REMARKS

FIG. 325.—Method of recording the results of fuel-pump tests.

and read off a pressure gauge provided for this purpose. The principle of the nozzle tester is to apply a similar value of the fuel pressure to that given by the injection pump and to read off the pressure on a suitably calibrated pressure gauge. The nature of the nozzle spray can then be studied under these given pressure conditions. Fig. 326 illustrates the C.A.V. nozzle tester in which the fuel is fed to the nozzle holder from the tank, 1, through a stopcock and the pressure pump, 4, which is operated by the handle, 5. The pressure gauge, 6, which is connected to the pressure pipe, 7, reads up to 300 atmospheres (4410 lb. per sq. in.). The shut-off cock, 2, is provided so that the gauge may be shut off during nozzle tests. When the nozzle tester is to be used it must first be cleared of air. To

do this the air release screw, 3, is removed. When fuel oil has flowed out for about 3 seconds the screw is replaced and tightened up. The hand pump is then used until fuel flows from the pressure pipe, 7. The apparatus is then ready for use and the nozzle holder with its nozzle in position can be connected up. The nozzle pressure is adjusted by means of the adjusting screw, 8, after releasing the locking nut, 9 ; the value of this pressure is than read off the gauge.

Apart from its other uses the nozzle tester can be employed for testing nozzles removed from engines, for various possible defects or for readjustment purposes.

The N.A.C.A. Compression-Ignition Test Engine Unit.—In connection with various researches that have been made at the Langley Memorial Aeronautical Laboratory on fuel spray and combustion problems occurring with compression-ignition engines, a special engine test unit was evolved and has since been employed with additional appliances to a number of investigations forming the subjects of Reports[1] issued by the American National Advisory Committee for Aeronautics.

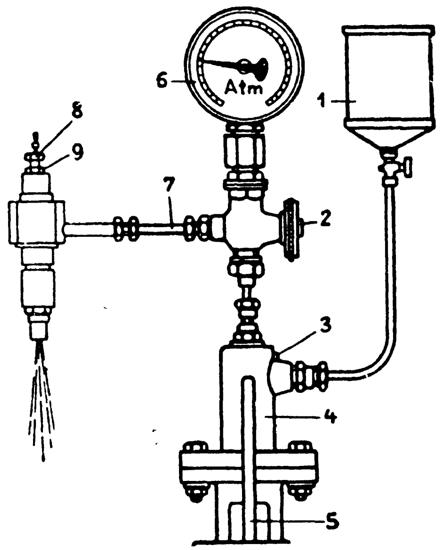

FIG. 326.—Injection nozzle testing apparatus (C.A.V.).

The apparatus, shown schematically in Fig. 327, and in some detail in Fig. 328, consists of a single-cylinder engine unit of 5-inch bore and 7-inch stroke, the compression ratios, which were different for the various tests, forming the subjects of the Reports, ranging from about 13 to 17 to 1.

The apparatus shown in Fig. 328 was used in connection with the tests mentioned in the footnote reference, Report No. 525, and employed a different injection system to that shown in Fig. 327 ; the original injection system was used to operate an automatic scavenging or compression release valve.

[1] Three Reports of particular interest are as follows :

" The N.A.C.A. Apparatus for Studying the Formation and Combustion of Fuel Sprays," A. M. Rothrock, Report No. 429, 1932.

" Fuel Vaporization and its Effect on Combustion in a High-Speed Compression-Ignition Engine," A. M. Rothrock and C. D. Waldron, Report No. 435, 1932.

" Some Effects of Injection Angle Advance, Engine Jacket Temperature and Speed on Combustion in a Compression-Ignition Engine," A. M. Rothrock and C. D. Waldron, Report No. 525, 1935.

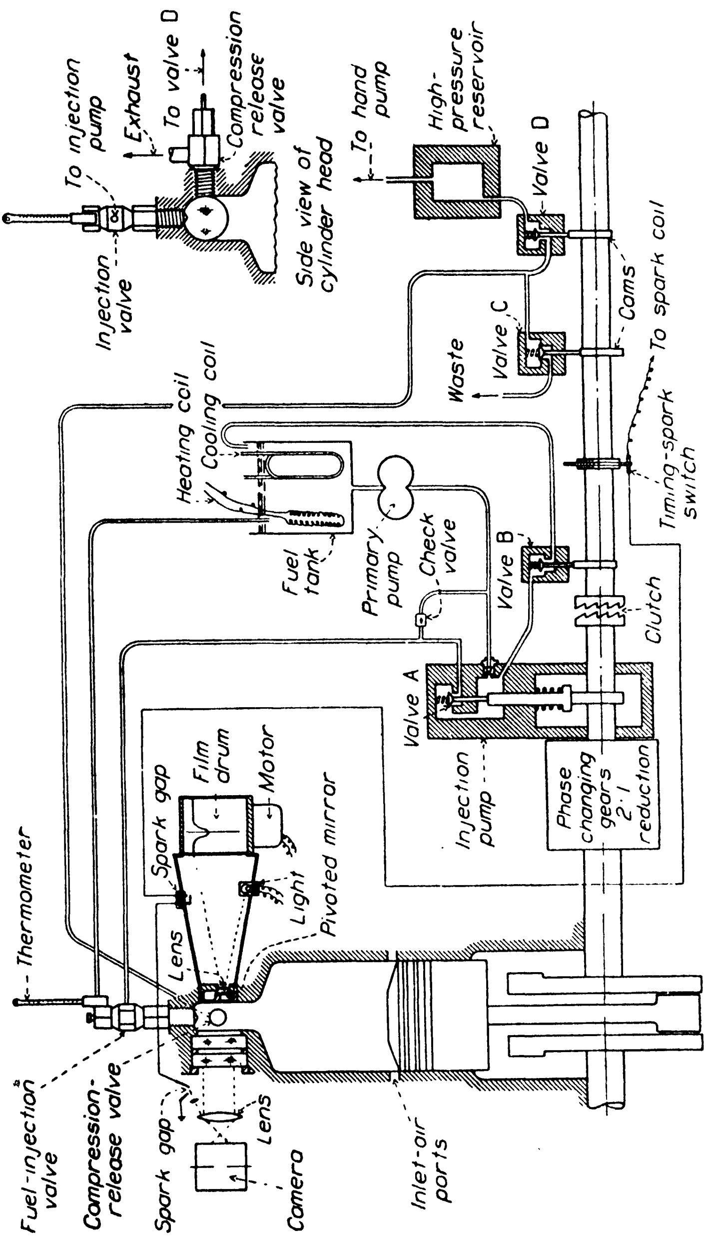

FIG. 327.—Diagrammatic layout of N.A.C.A. compression-ignition engine combustion testing apparatus.

Referring to the arrangement shown in Fig. 328, this consists of the single-cylinder test engine previously mentioned, an electric motor for driving the test engine, a fuel-injection system driven from the crankshaft of the engine, and a high-speed photographic system.

The combustion chamber of the engine has a diameter of 3 inches and a depth of $\frac{7}{8}$ inch. This shape was chosen because it permits the two sides of the chamber to be made of glass discs. There are two 1-inch thick windows on each side of the chamber separated by an air space which is connected to a tank of compressed air. Since the air temperatures of 1100° to 1700° C. absolute and pressures in excess of 800 lb. per sq. in. are reached in the combustion chamber, the conditions to which the inner windows are exposed are extremely severe. The maximum stress on the inner windows is reduced by maintaining an air pressure between the windows of approximately 450 lb. per sq. in. The combustion chamber is connected to the displacement volume of the engine by a rectangular orifice of a size (0·695 sq. in. in area) to produce calculated air velocities of 300 ft. per sec. in the chamber.

There are two openings in the cylinder head for the injection valve so that the effect of air velocities can be studied with the spray directed normal to or counter to the air-flow. The third opening is used for a maximum pressure indicator.

At the bottom of the stroke the piston uncovers ports in the cylinder wall. These ports are connected to a cam-operated poppet valve so adjusted that it is open when the piston is at bottom centre. These ports and the valve permit air to enter the cylinder and compensate for air leakage around the piston rings. In addition, the inlet manifold may be connected to an air compressor so that the effect of increased air density on the fuel spray and on the combustion may be studied.

The cored passages in the cylinder head and the jacket around the cylinder are connected to an electrically heated tank containing glycerine. By means of this liquid, temperatures of 260° C. can be maintained in the cylinder jacket and the cylinder head. A suitable pump is used to circulate the glycerine.

One end of the crankshaft is connected to the electric driving motor, and the other end to the timing gear through which the injection system is driven. The timing gear is calibrated so that the start of injection can be varied in increments of one crankshaft degree. The shaft connecting the timing gear to the injection system is separated by a clutch, *d*, similar to those employed on press punches. When this clutch is engaged by means of the mechanism shown in Fig. 328, the camshaft of the injection system runs at one-half engine speed.

The injection system employed consists of a high-pressure

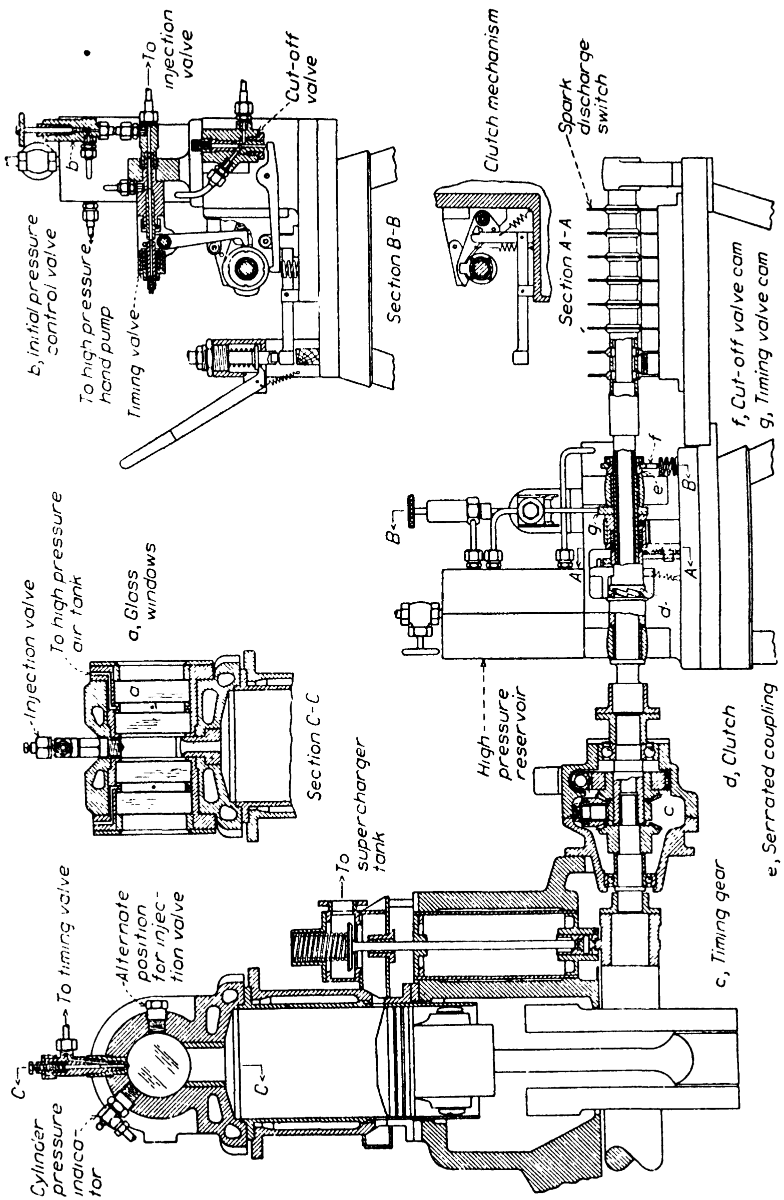

FIG. 328.—N.A.C.A. compression-ignition test engine unit.

reservoir to which fuel is forced under pressures up to 10,000 lb. per sq. in. by means of a hand pump; a timing valve connected by suitable tubing to the injection valve; a by-pass valve for controlling the injection period; and a valve for controlling the initial pressure in the injection tube before the start of injection

The tube connecting the timing valve and the injection valve is 50 inches long so that the instantaneous pressures at the discharge orifice will not fluctuate because of the pressure-wave phenomena. A hand-operated needle valve in the top of the high-pressure reservoir allows air to be released from the reservoir.

When the clutch mechanism is engaged by the operating lever the first cam opens the timing valve, which releases the fuel under pressure in the reservoir to the automatic injection valve. Injection continues until the second cam opens the by-pass valve, at which time the hydraulic pressure in the high-pressure reservoir is released to atmospheric pressure and injection is stopped. The period of injection can be varied by means of the serrated coupling connecting the by-pass valve cam to the camshaft.

The test plant was provided with a high-speed photographic apparatus. The cylinder head of the engine has a vertical disc form of combustion chamber provided with glass window sides. When the fuel is injected into the combustion chamber cinematograph film pictures at the rate of 2000 per sec. are taken of the fuel spray formation by means of spark discharges. When combustion takes place the light of the combustion is recorded on the same film as the spray photographs. The arrangement of the original photographic apparatus is shown in Fig. 329. The Reports previously referred to contain numerous photographs of fuel sprays and combustion phases.

Referring to Fig. 327, which shows the slightly modified N.A.C.A. combustion apparatus, the high-speed cinematograph camera employed permitted the flame formation to be studied by means of individual photographs instead of in one continuous photograph for the whole cycle, as in the previous tests. In reference to the modified injection system, the compression-release valve is controlled by the valves shown at C and D and the high pressure reservoir. With this arrangement full compression exists only for the single cycle in which the injection of the fuel occurs, the engine being scavenged on all preceding and succeeding cycles.

The apparatus is brought to speed by the electric driving motor. During this time, on each stroke of the piston the air is forced out through the compression-release valve in the cylinder head. On the down strokes the compression-release valve closes, because of the reversal of air-flow, and a partial vacuum is created. When the piston uncovers the ports at the bottom of the stroke, fresh air is inducted into the displacement volume.

When the test speed is reached, the clutch is engaged for a single revolution of the camshaft, that is, for two revolutions of the crankshaft. When valve B closes, the plunger in the injection pump compresses the fuel in the small pump reservoir. This compression of the fuel holds the valve stem A against its seat so that

none of the fuel can enter the injection tube from the small reservoir. Shortly before the pump plunger reaches the top of its stroke it engages the valve stem A, causing the stem to lift from its seat.

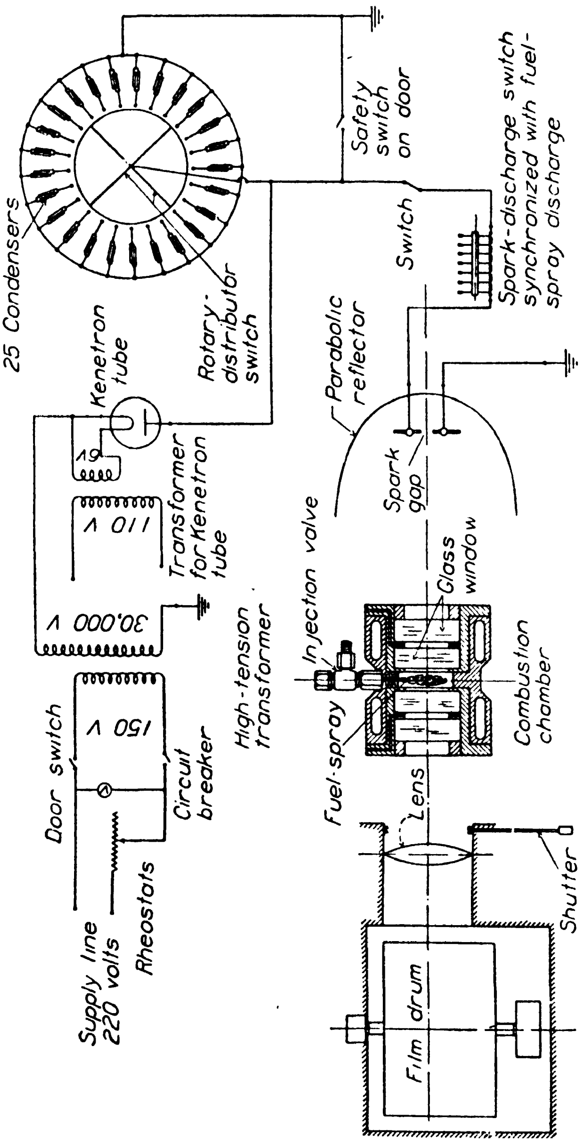

FIG. 329.—Arrangement of N.A.C.A. photographic apparatus for combustion tests.

A hydraulic pressure wave of high intensity is transmitted through the injection tube to the injection valve. Injection continues until the force exerted by the wave drops to a value less than the closing pressure of the injection valve. The fuel quantity discharged is

controlled by the injection valve opening pressure. The temperature of the fuel in the injection valve is maintained at a constant value by continuously circulating fuel from the fuel tank, through the injection tube, through four 0·010-inch holes to the centre of the hollow injection valve stem, and back to the fuel tank.

Photographic Apparatus and Indicator.—In most of the tests a glass window was installed in one side of the combustion chamber, and the optical indicator in the other side. A synchronous motor drove the indicator film drum at a constant peripheral speed of 100 inches per second. Because of the large stiff diaphragm in the indicator, the optical record closely followed the course of the pressure rise within the combustion chamber except in the case of detonation, which set the unit vibrating.

The camera used in these tests permitted high-speed motion pictures to be taken of the flame spread in the portion of the combustion chamber directly behind the 2½-inch glass window. This camera takes motion pictures at rates up to 2250 frames per second by the use of a prism rotating at a high speed. The exposure time is one-third the time interval between the exposures. In order to synchronize the motion pictures of the flame spread with the indicator card, two sparks 90 crankshaft degrees apart were simultaneously recorded on the two records.

In another series of tests the indicator was removed and glass windows were installed in both sides of the combustion chamber. A 1000-watt light was then directed through a ground glass on to the window that replaced the indicator. When the motion pictures were obtained with this set-up, a silhouette of the spray was recorded before the start of combustion. The intensity of the subsequent combustion was sufficiently greater than the intensity of the light from the bulb that the combustion was recorded on the film.

CHAPTER XII

THE TESTING OF AIRCRAFT ENGINES

AIRCRAFT engines intended for installation in aircraft as distinct from experimental or development models, are required to undergo a number of searching tests before they are accepted for service in the aircraft. These tests are designed to show the general reliability, endurance qualities and standard of performance of the engines; they include in general tests for endurance over a stipulated period of time, measurements of output, speed, fuel and oil consumption under part and full throttle conditions, starting, slow-running and acceleration tests, etc.

Each country has its standard tests in connection with the International Air Navigation Regulations. In this country the requirements for engines intended for use in civil aircraft are set forth in the Air Ministry Publications No. 840, with subsequent addenda, and No. 1208, Design Leaflets C1, C2, and C3.[1]

As all of these tests involve the running of the engine under load a special form of power absorption or transmission brake (or dynamometer) provided with means for measuring the engine's power output at various speeds, is employed in the test-house. Formerly, it was not uncommon to fit the engine with a calibrated airscrew or paddle-type dynamometer and to test it for output, reliability and endurance. The principal dynamometers used for high-power aircraft engines are the hydraulic absorption and the electric generator types, previously described.

In the case of the larger aircraft engines the combined hydraulic and electric dynamometers previously referred to in Chapter VI are often used.

Before proceeding to give an outline of modern aircraft engine test procedure, it may be of interest to describe the general design and equipment of the test-houses that have been scientifically designed in order to overcome the difficulties experienced in the earlier constructions.

Aircraft Engine Test-houses.—In view of the large power outputs of modern aircraft engines, namely, from 400 to over 2000 h.p., the design of the test-house requires very careful consideration from the point of view of reducing the volume of the exhaust noise to a minimum whilst permitting the exacting tests required by official

[1] Subsequent amendments, made during the war of 1939, had not been made public in 1942 but are available to aircraft engine manufacturers on Government contracts.

specifications to be carried out in a convenient manner by the personnel.

In this connection it may be of interest to describe the test-houses designed by the Bristol Aeroplane Company for testing their engines. These buildings are arranged in groups of two dynamometer test-houses, supplemented by a number of units of generally similar design, but containing test-stands of the cable-suspended type, for tests on engines fitted with controllable pitch airscrews.

The principal requirements concerned in the design of these buildings are as follows :—

(1) Improved working conditions for the test personnel.

(2) Noise to be reduced to a minimum, both in the control room and in the surroundings of the test-houses, by means of an efficient sound-absorption system, dealing with the whole range of frequencies existing under all the various engine test conditions.

(3) Simplified operation, providing practicable one-man control for each engine.

(4) The use of fireproof materials in the construction of the engine chambers and exhaust tunnels. These materials were also required to be of a type which would not be fouled or spoiled by oil saturation.

Sound Absorption.—As a result of preliminary acoustic experiments made with the aid of special N.P.L. equipment on an old test-house, the intensities of the noises ranging in frequency from 36 to 9000 cycles per second were measured at a number of different positions and the results analysed. The data obtained gave valuable guidance in the arrangement of suitable sound-absorption devices in the new buildings.

A variety of different damping materials, such as slag-wool and compressed, fire-proofed wood-shavings, is used, and the effectiveness of the scheme is very largely due to the way in which these materials have each been used to best advantage in dealing with a particular range and location of sound-frequency. Another very important factor which has been taken into account is the necessity of preventing structure-borne sound from reaching the outside walls of the test-houses.

The degree of silencing achieved by this practical application of modern acoustic theory is most impressive to anyone who is familiar with aero-engine testing under previous conditions. In the control room, with two engines developing a total of over 2000 h.p. only a few feet away, conversations may be carried out as comfortably as in any normal works office. In addition to the absence of noise as generally understood, there is also a complete elimination of that pulsation in the air which is often a particularly unpleasant feature in the neighbourhood of a large power test-plant.

From the standpoint of the rest of the factory, the effect is

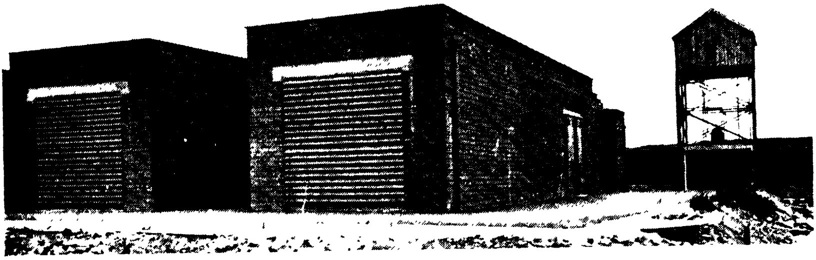

FIG. 330.—General view of the Bristol Aeroplane Company's silenced test-houses.

[See page 375.

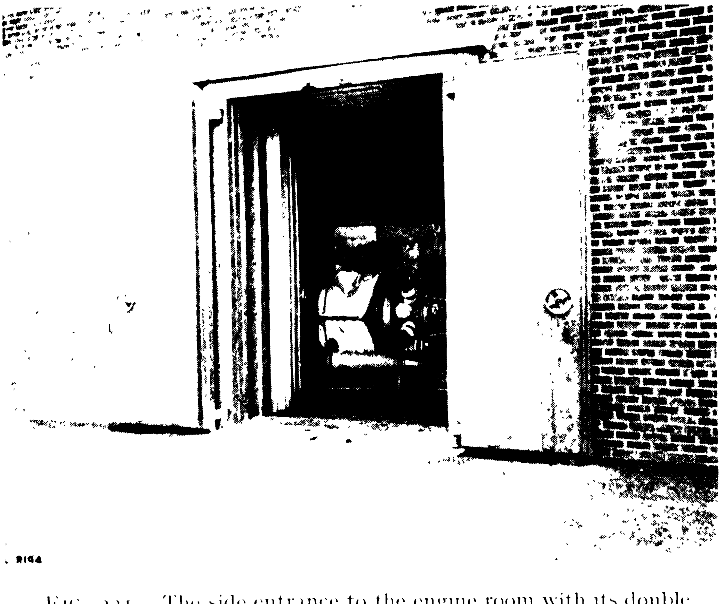

FIG. 331.—The side entrance to the engine room with its double sound-proofed doors.

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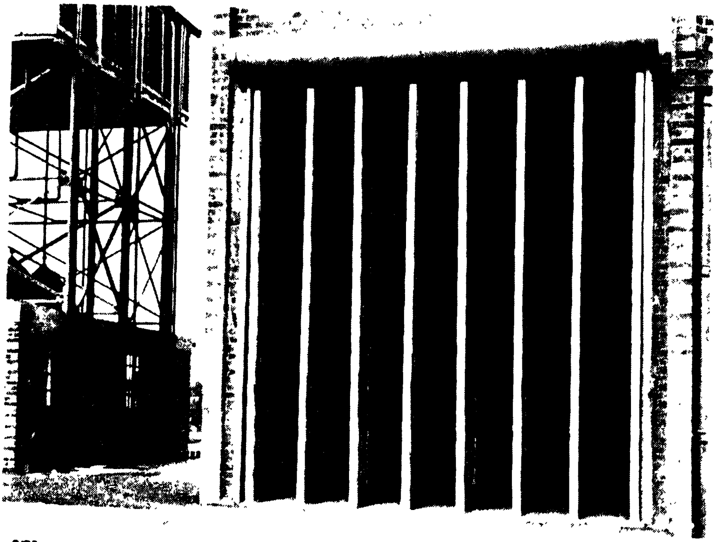

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FIG. 332. Exit of exhaust tunnel showing the specially shaped sound absorbing panels.

See page 376.

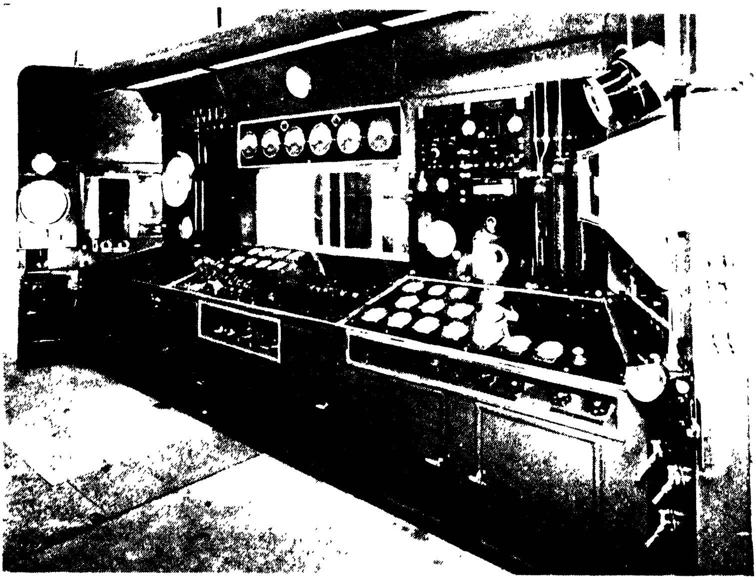


FIG. 333. Control panel of Froude aircraft engine wind tunnel test plant. (Messrs. Armstrong-Siddeley Motors, Ltd.)

[To face page 377.

likewise excellent in its relief to the ears and nerves. Even immediately outside the test-house, again with two engines running at full power within, only a very subdued drone can be heard, and a few yards away from the building even this murmur becomes inaudible above the sounds which are normal in any industrial locality.

In more scientific terms, it may be said that a noise of about 138 phons equivalent loudness exists near an engine being tested on the ordinary " unsilenced " type of test-bed. This noise is reduced by nearly 50 per cent. in the control room of one of the new test-houses, and in the zone immediately outside the building.

General Construction.—Each of the double test-houses has its own control room centrally located between the two buildings, and the latter communicate at one end with the cooling-air intake tunnel and fan chamber, and at the other end with the exhaust tunnel.

The engine room has inner and outer walls which are entirely independent of each other and rest on separate foundations. Both sets of walls are of the " cavity " type, so that there are four thicknesses of brickwork and three air spaces between the interior and the outer atmosphere. The reinforced concrete roof is carried by the inner walls and does not make contact with the outer ones. The sound-absorbing panels which line the inner walls are metal-faced and easily cleaned. The engine bed is isolated from the floor, and the latter is isolated from the walls.

The cooling-air tunnel and fan chamber are of solid construction. The outer face of the intake has sound-proofed metal louvres to prevent the ingress of rain or other foreign matter. For a certain distance the intake tunnel is fitted with sound-absorbing panels to prevent sound travelling back through it to the outside air.

The exhaust tunnel is built in a similar manner, but is much longer. The first part of its length, from the engine room, consists of a sound-proofed metal honeycomb. The remainder is fitted with a carefully designed labyrinth of sound-absorbing panels, which are quickly detachable and easily cleaned.

The arrangement of the cooling-air fans, water-brake dynamometers, and engine-beds, is generally similar to that already in use in the older test-houses. The load capacity of all items has, however, been suitably increased to provide for testing the much more powerful engines now concerned. The dynamometers also have water sluices operated by electric motors instead of by hand, thus facilitating a remote-control system.

The cooling-plant for the dynamometer brake-water circuit is conveniently housed between the two tunnels in a separate room which communicates with the control room. An efficient water-softener is included, to guard against lime deposits in the brakes or pipe-lines.

Engines are admitted to the engine rooms directly, through side entrances which are fitted with double doors of heavy, sound-proofed, steel construction.

Control Room Equipment.—The control room (Figs. 334 and 335) is equipped with two main control units, one for each engine room. Each unit carries all levers and instruments for the operation and observation of the engine concerned, and is designed for one-man control. The equipment in each case includes :—

Throttle, mixture and slow-running cut-out controls.

Five oil thermometers (for main engine feed, carburettor jacket, sump, scavenge outlet, and tank).

Two oil pressure gauges (for main engine feed and oil cooler).

Two air thermometers (for air tunnel and carburettor air intake).

Four water thermometers (for dynamometer brake water circuit).

Two fuel pressure gauges (for engine fuel pump).

Two tachometers, one showing airscrew shaft speed and the other crankshaft speed. The latter instrument is fitted with a relay-switch device which breaks the ignition circuit should the speed tend to rise above a predetermined limit, thus preventing accidental damage to the engine due to excessive revolutions.

Automatic devices are also included in the water-brake circuit and in the air-tunnel, to switch off the engine should either of these ancillary services fail.

Visual fuel-flow bowl, for observing fuel circulation through engine fuel pumps when these are not being used to supply the carburettor.

Four fuel flowmeters, fitted with "zero-head" float control and adjustable for height to suit engines fitted with either up-draught or down-draught carburettors.

Two boost pressure gauges, one for measuring the pressure at the carburettor intake when using the depression chamber for obtaining power curves under "altitude" conditions; and the other for measuring the induction pressure in the engine. These gauges are of the mercury type and are calibrated to give direct readings, corrected for barometric pressure, in lb. per square inch, thus saving considerable time in taking observations.

It is important to note that no controls or pipe-lines are anywhere in contact with the outer walls of the engine chambers. They are taken through isolated channels below the floor, so that they cannot transmit sound-vibrations to the outer walls and thence to the atmosphere.

Another interesting feature is that all oil consumption and circulation data is obtained by weight and not by volume. Errors due to the presence of oil-froth are thus avoided. The oil-weighing machines and tanks are located immediately adjacent to the control-boards.

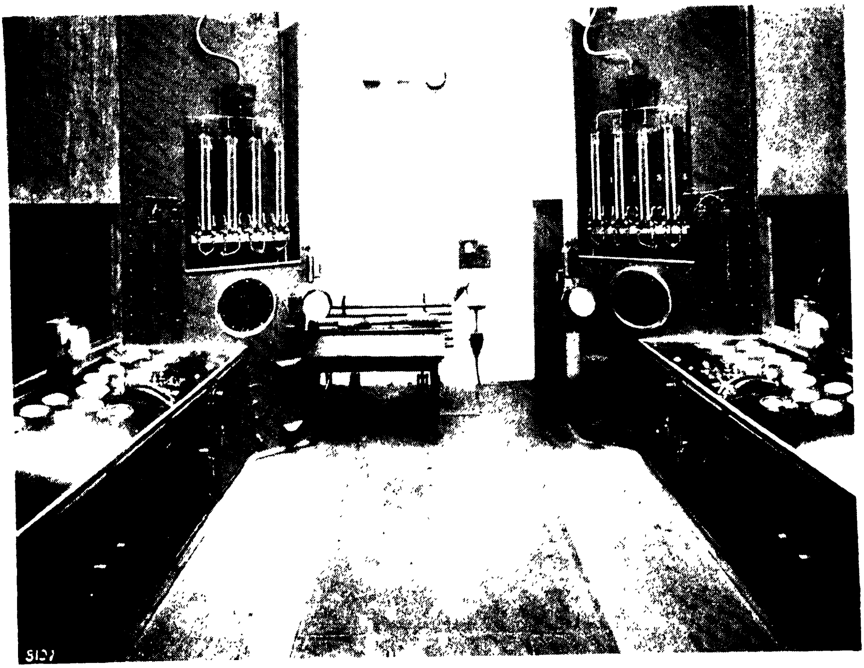
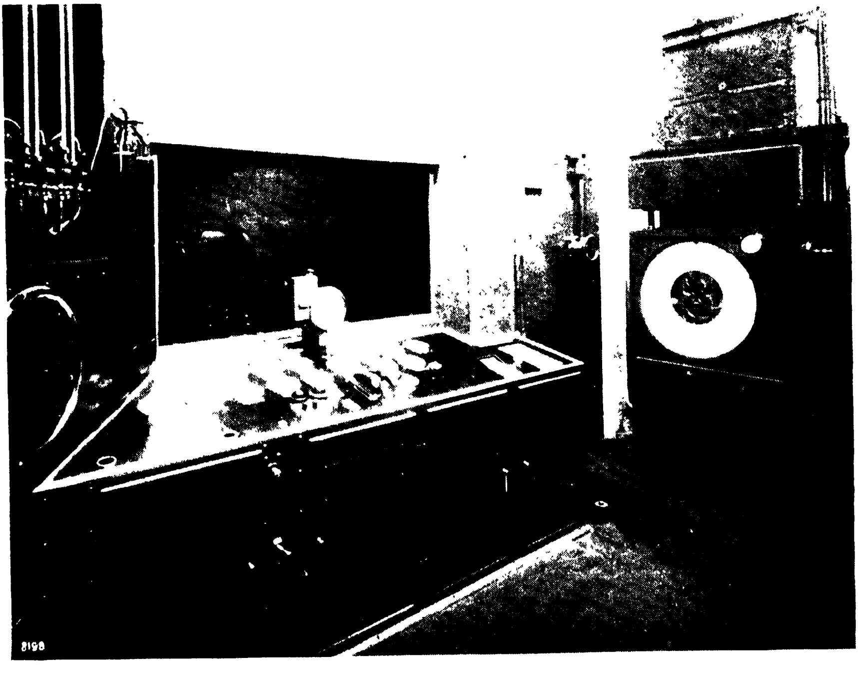


FIG. 334.—General view of control room, showing the duplicated control boards, fuel flowmeters, boost gauges, etc. (*Bristol Aeroplane Company.*)



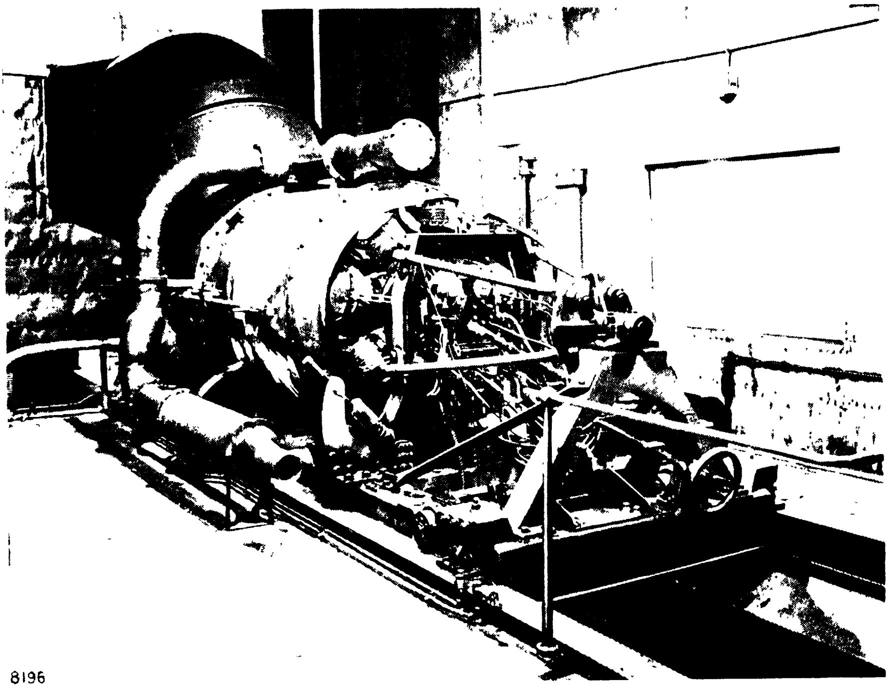
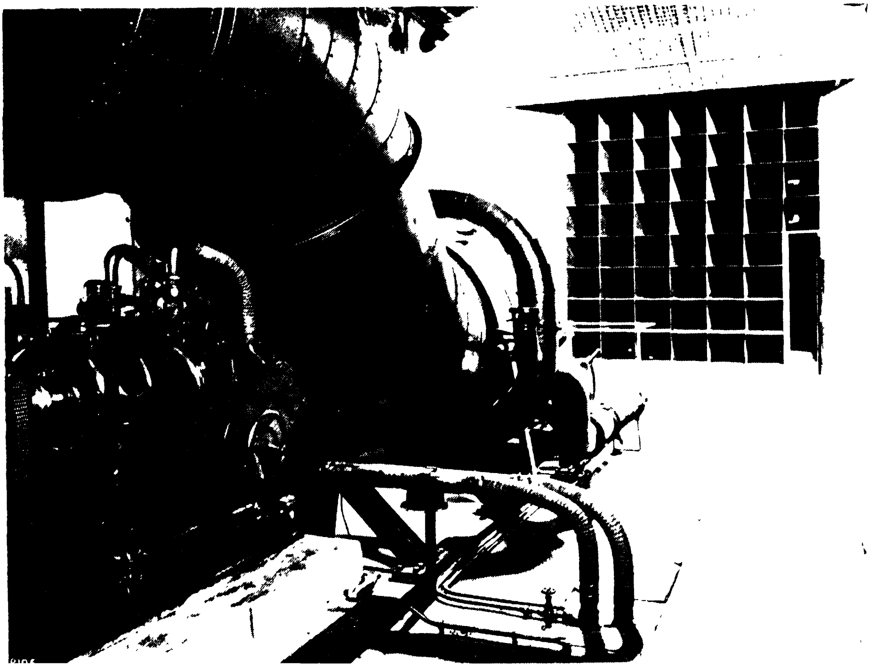


FIG 337 Interior of one engine room The fan chamber is beyond the partition on the left The triple glass window to the control room is seen on the right



The engine under test is watched by the observer through a window immediately above the control-board (Fig. 335). This window is fitted with armoured triple glass, there being an air space between each pane. The engine rooms are illuminated by powerful incandescent-vapour type electric lamps; there are no outside windows. Double doors, similar to those already mentioned, give access from the control room to the engine rooms.

Outside the main test building is a large underground fuel tank, surmounted by a pump room. The fuel is raised by electrically driven pumps to an elevated tank, providing gravity feed to the zero-head float chambers of the fuel flowmeters in the control room.

The engine-and-airscrew test-houses which are in course of erection, employ generally similar principles of layout and sound-absorption to those already described.

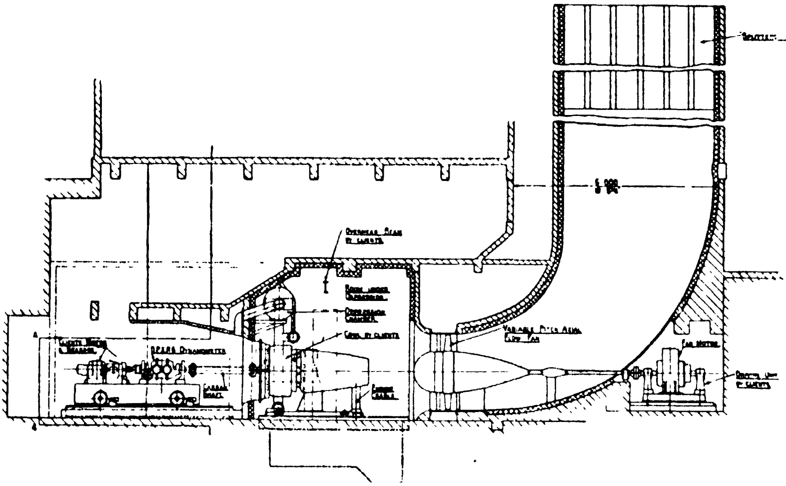


FIG. 336.—A typical test installation. (*Heenan and Froude.*)

A Typical Test Installation.—Fig. 336[1] shows a test-house and equipment for aircraft engine tests. The engine in this case is mounted upon a stream-lined nacelle, somewhat similar to that which is used in flight. To simulate flight conditions, the engine is fitted with a cowl. On the test plant the effect of attaching the cowl, given an efficient design, is to increase the velocity of the air flowing over the cylinders without necessitating the greatly increased fan power which would otherwise be necessary. The variable speed fan sucks air from the chamber in which the engine is mounted. The velocity of the air, in the vicinity of the cylinders, approaches 200 miles per hour. The power of the engine is transmitted through a long tubular cardan shaft to a reversible

[1] "Testing Methods. Brakes and Dynamometers," G. H. Walker, *Coventry Eng. Soc.*, 1938.

"Froude" dynamometer having motor-operated sluice gear, while the engine can be motored and started by an electric motor of 150 h.p., a gear-box and self-disengaging jaw coupling. Current practice is to replace the jaw coupling with a freewheel clutch combined with jaw clutch, which greatly facilitates the starting of refractory engines.

A "Heenan" mechanical water cooler and circulating pump receive the hot water discharged by the dynamometer and after cooling return it under pressure to the dynamometer.

The operators are protected from the noise of the engines by a sound-insulated cabin fitted up with a very complete system of remote controls, instruments, gauges, etc., from which the engine power, altitude conditions, dynamometer load, airblast speed, and many other features can be controlled and measured.

The complete test-house is sound-insulated so that external noise is reduced to a minimum.

General Testing Procedure.—When the new engine has passed inspection it is mounted on a suitable engine stand and given *a running-in operation* for the purpose of bedding down the main big-end, camshaft and other bearings, the pistons and cylinder walls, etc.

For this test the engine is motored electrically with the ignition and fuel supply to the carburettor switched off. The running-in speed is usually from 500 to 600 r.p.m. and occupies about five hours. During this operation the engine is liberally lubricated with a special low viscosity oil which, during its circulation, is passed through a centrifuger of the de Laval pattern to remove any solid deposits. After the running-in process the engine is transferred to the dynamometer engine stand and connected to the dynamometer by means of a flexible coupling. The subsequent test procedure can best be described by referring to the general practice adopted for Bristol air-cooled radial engines. The testing arrangements are necessarily somewhat elaborate in view of the number and importance of the exacting test standards adopted for these engines.

Fig. 337 illustrates one of the Bristol engine test-rooms, and shows the large cooling air duct surrounding the engine seen under test. The fan chamber for this duct is beyond the partition on the left. On the right can be seen the large triple-glass window separating the test-room from the control-room in which the test operators work. Fig. 338 shows the Froude hydraulic dynamometer, and, on the right, the honeycomb entry to the sound-absorbing exhaust tunnel.

The large air ducts enable cooling air speeds up to 180 m.p.h. to be employed, although normally a speed of 100 to 120 m.p.h. is used; in order to obtain the required volume of air and cooling speeds a 500 h.p. electric motor is used to drive the fan.

The engine is mounted on a movable cradle for facility of coupling up with the dynamometer; the latter—as shown in Fig. 338—is situated at the rear of the air duct; at the back of the dynamometer there is a 100 h.p. electric motor for the purpose of motoring and starting the engine under test (Fig. 339).

The Oil Supply System.—The test stand has a *duplicate oil supply system* to enable either castor-blend or mineral oils to be used during the tests. The former grade of oil is employed for initial running of the engine under its own power, after which the normal mineral oil is used. The oil supply tanks are provided with electric heating elements and water-cooling coils to vary the oil temperature as required; usually it is held at about 70° C.

The fuel supply is connected to three flowmeters for measuring the fuel consumption, and thence to the carburettor on the engine. The flowmeters are also used in connection with the adjustment of the carburettor jets.

Other Test Equipment.—In addition to the usual carburettor controls the equipment of the test-stand includes twin ignition switches, two tachometers and a means of checking the speed with a hand rev.-counter, oil pressure and temperature gauges, air speed indicator, thermo-couples for measuring the cylinder and other temperatures, boost pressure gauge, U-tubes and depression chambers for the testing of supercharged engines.

Preliminary Running Under Power.—When the various pipes and controls have been connected to the engine the latter is motored around and the cooling fan started. If everything is in order the ignition is switched on and the engine is allowed to run light at about 800 r.p.m. for a period of half an hour. The load is then applied gradually and the throttle opened until by the end of a further half-hour the engine is developing its proper test load. In the case of *supercharged engines* the load is 90 per cent. of the ground level power at normal r.p.m. and with the rated boost pressure in the induction system. For *non-supercharged* engines of the high compression type the load is 90 per cent. of the rated ground power level.

During the tests flowmeter readings are checked and any necessary adjustments made to the carburettor to give the correct fuel consumption and power. Afterwards the engine is run for one and a half hours at 90 per cent. load and at normal speed.

Up to this stage of the test the vegetable base lubricating oil has been used. This is now drained out, the filters examined, and the oil system changed over to the officially specified mineral oil, e.g. D.T.D. 109, and the engine is opened up for a further hour at 90 per cent. load.

During the last five minutes of this hour unsupercharged engines are opened up to full throttle, using a leaded fuel in the case of high

compression engines or up to normal ground level power for supercharged engines.

Type Tests for New Designs of Engines.—The first engine of a new design or type before being accepted as "airworthy" by the Air Ministry, has to undergo an official type test before which the manufacturers must declare its proposed power rating.

The following is an extract from Air Ministry Publication 840, hitherto used for type tests for new designs of engines:—

Preliminary.—Before an engine is submitted for type test, the maker must have declared the rated full power, normal and maximum permissible r.p.m., etc. In addition, the engine should have satisfactorily completed the standard two hours' endurance test; have been stripped for examination, measurement and recording of engine data such as bore, stroke, compression ratio, weight, etc., and finally re-assembled and tuned up on the test-bench. The type test should be started with the taking of a power curve.

Endurance Test.—An endurance test of 100 hours' duration at normal speed at 90 per cent. full power (unless otherwise specified) is to be taken. This endurance test will comprise ten non-stop runs of ten hours' duration. Fifty hours of this test will be carried out with the engine fitted with an airscrew, such airscrew being suitable for the thrust test. The last ten hours' run must be carried out on an approved form of brake, and at the commencement of the one-hundredth hour of the endurance test the load will be increased so that the engine is running at full power at normal r.p.m. until within five minutes of the completion of the test, when the engine will be opened out to full throttle.

For engines intended for civil aircraft the duration of the endurance test shall be fifty hours, to comprise five non-stop runs of ten hours' duration. Twenty hours of this test will be carried out with an airscrew. The load will be increased to full power on the fiftieth hour, otherwise the conditions will be the same as laid down in the preceding paragraph.

Slow Running and Acceleration.—The engine must be run as described on page 384. The duration of the slow running is to be ten minutes.

High Speed.—The engine to be run for one hour continuously at 5 per cent. in excess of the established maximum permissible speed, and under load conditions at the option of the manufacturer.

High Power.—The engine is to be run for one hour continuously at the established maximum permissible speed and at full throttle.

Power Curve.—Another power curve to be taken.

Dismantling.—On completion of the above tests the engine is to be completely dismantled for inspection by the A.I.D. inspector. On completion of inspection, any defective or unduly worn parts

that have been rejected must be renewed and the engine re-assembled for test.

Final Test.—If the condition of the engine has been satisfactory, it will be required to be submitted to a final test of thirty minutes under the same conditions as the endurance test.

During the preceding tests the consumptions of fuel and oil are measured on any non-stop run of ten hours' running of the fifty hours' duration test, and these consumptions, in the case of civil aircraft engines, must be within 20 per cent. of the makers' rating. The actual fuel consumptions laid down depend upon the compression ratio and grade of fuel employed; the consumptions are usually specified for the type of engine in the official specifications. In regard to the *oil consumption*, this should not exceed 0·025 pint per B.H.P. hour for water-cooled engines and 0·045 pint for air-cooled ones.

The *water pump delivery* is also required to fulfil certain conditions: at normal r.p.m. the engine water pump will be required to deliver at least 15 gallons per minute per 100 B.H.P. (normal) against a circuit resistance or head of 2 lb. per sq. in. in excess of that required to overcome the hydraulic resistance of the engine, while the water just before the pump inlet branch is maintained at a temperature of not less than 75° C., and its pressure at least 4 lb. per sq. in. below atmospheric. This test will be carried out with the engine developing the rated full power at normal r.p.m. for at least thirty minutes. At the slowest speed at which the engine will run fitted with the airscrew used during the thrust test, the pump must be capable of circulating water through the engine with a resistance at the water outlet branch equivalent of 2 feet head of water.

In connection with the *maximum speed tests*, Bristol engines are tested by the makers at 15 per cent. over the normal speed instead of the usual 10 per cent.; and at 5 per cent. in excess of the maximum rated speed on a moderate load.

The Engine Accessories.—In regard to the *endurance test*, this is arranged to cover not only the performance of the engine itself, but also of all of its auxiliary parts such as the fuel pumps, carburettors, magnetos and other standard accessories supplied with the engine.

Final Inspection.—As previously noted, after the type tests have been completed the engine is stripped for inspection by the A.I.D. representative, the parts having first been washed and cleaned. Any corrections or replacements necessary are made and the engine is then re-assembled and taken to the test-house, where it is mounted on a dynamometer stand and run with mineral oil lubricant for half an hour, the load being increased gradually to the 90 per cent. figure as in the case of the endurance test; this load

is held for one hour, the last five minutes as before being at full throttle or rated boost.

The final test employed for Bristol engines, in addition to the running already described, includes a power curve, a throttle curve, and, in the case of supercharged and altitude-rated engines *an altitude curve* and a *constant boost curve.*

In connection with the *slow running and acceleration tests*, the engine, after its endurance tests, is generally taken to an open hangar and mounted on its test-stand. The power is absorbed by a four-bladed test airscrew which provides also the necessary cooling air stream. Each stand is operated from a cabin and the equipment as regards controls, fuel and oil circulation, instruments, etc., is similar to that of the dynamometer test-house. To satisfy the official requirements engines should run reasonably slowly and be able to open up to the normal r.p.m. within five seconds without any excessive " popping " or rough periods of running ; they should run reasonably well at 80 per cent. reduction from normal r.p.m. when the dynamometer has been adjusted for 90 per cent. power at normal r.p.m.

Thrust Tests.—In order to test the thrust bearings of the engine it is usual to submit these bearings to a thrust of 6 lb. per B.H.P. on the airscrew shaft during tests made under the same conditions as the endurance tests ; if, however, a suitable airscrew is fitted the thrust test can be made with this. The direction of the thrust should be the same as that of the airscrew normally fitted in the aircraft in which the engine is to be installed, i.e. tractor or pusher.

Tilting Tests.—Engines intended for military and similar purposes where severe climbing, banking, and diving conditions have to be provided for are usually given independent tests on a tilting test-bed capable of holding the engine rigidly at any given angle to the horizontal, up to the vertical diving attitude. In this way the fuel and oil systems can be tested and the other components likely to be affected by the inclination of the machine—as, for example, the controls and cables—checked.

Fig. 340 shows one of the tilting test-stands used by the Bristol Company, with a " Pegasus " engine undergoing a 70-degree tilting test. In this type of test-stand all controls and feed lines with the exception of the lubrication system—which is mounted on the trunnion arm and moves with the engine—are flexible in order that movement of the engine is unrestricted ; the controls are actuated by means of Arens cables.

Controllable Pitch Airscrews for Testing Engines.—In connection with the production tests of aircraft engines for endurance and similar purposes there are several advantages in using controllable pitch airscrews instead of dynamometers. Not only do

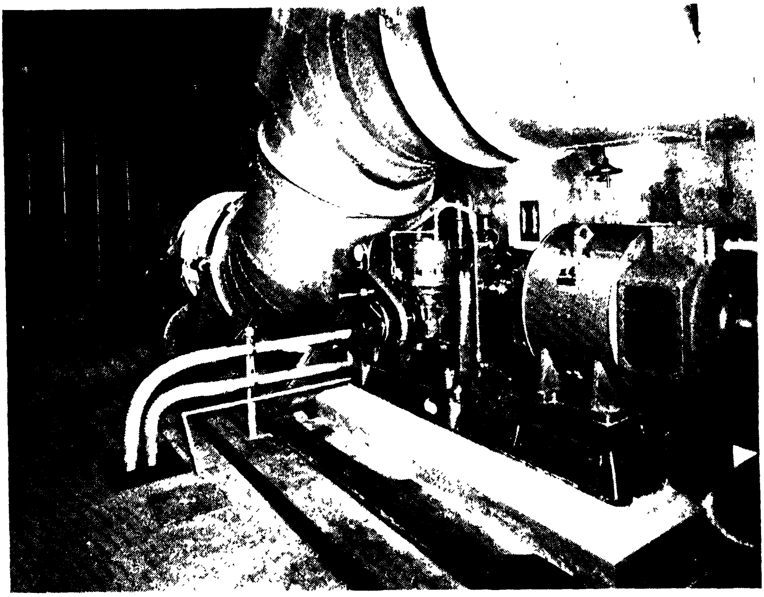


FIG. 339. Froude wind channel test plant in sound-proofed building, showing sound-absorbing exhaust tunnel at rear. (Heenan & Froude Ltd.)

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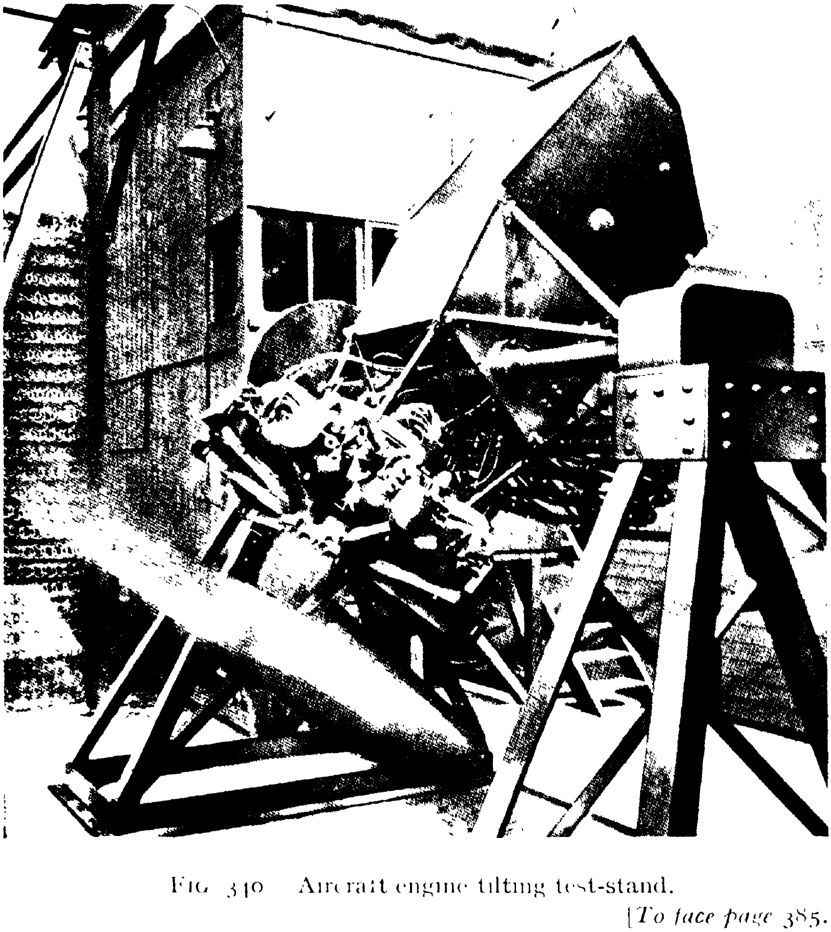

FIG. 340. Aircraft engine tilting test-stand.

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the operating conditions approach very closely those of flight but the engines are subjected to the same thrusts, lateral movements and torsional vibrations as in an actual aircraft, so that any inherent defects are exposed in this method, whereas they may not be apparent in ordinary dynamometer endurance tests. Again, airscrews are both cheaper, less bulky and more readily obtainable; they are also much easier to maintain. A further advantage lies in the absence of engine-dynamometer couplings that are sometimes the cause of trouble in engine tests and affect the torsional vibration frequencies. There is also the absence of great quantities of water to be cooled or electric energy to dissipate.

The Allinson aircraft engines are production tested in this manner, reinforced concrete test-stands with rubber mountings being used. It has been found necessary to provide an air straightener for the airscrew impelled air. This has been accomplished by using a plate having a bore slightly less than the airscrew diameter and locating it close to the trailing edge of the airscrew. There is no back-flow of air in test-stands equipped with this device and a greater air velocity through the cell is obtained.

Other Tests.—In the preceding outline of the usual test procedure and requirements for aircraft engines it has not been possible to include mention of any other special tests specified for military aircraft engines of recent design. Although these engines, in general, have to fulfil the test requirements previously described, other conditions may be specified by the Air Ministry and special instructions in regard to these issued to the manufacturers ; under peace time circumstances these instructions are available in the appropriate Air Ministry Publications to the public.

Measuring Engine Power in Flight.—Special types of transmission dynamometers are now available for use on aircraft to measure the engine output under flight conditions. A typical one is that developed at the R.A.E. on the basic principle of Ford's torsiometer.

The Bendemann hub dynamometer previously described is another device for measuring the engine power during flight.

A good résumé of the methods that have been employed for measuring the power output of aircraft engines in flight is given in a paper by N. S. Muir, read before the Royal Aeronautical Society (Feb. 25, 1937), from which the following abstract has been made:

The early instruments took their origin from the method of counterbalancing by weights and afterwards by the deflection of springs. Then the idea of measuring work by absorbing it in friction became known, and, finally, it was realized that to have some means of measuring power without absorbing it would be very desirable.

Dynamometers, therefore, may be divided into three classes, the gravity, the absorption, and the transmission types.

Reference is made in the paper to those important machines in each class which show the evolution and development of the art, and which we regard as being of historical value.

Smeaton, Prony, Hirn, Thomson, Froude, Osborne-Reynolds, Poncelet, Morin, Denny and Johnson, Moullin, and Ford have all contributed some important feature to the present state of knowledge of this art.

A brief reference to the problem concerning the variation of engine power with height and some notes on a few attempts at a solution by aerodynamic methods and by direct measurement are given in the paper.

A description is given of a new transmission dynamometer developed primarily for use in aircraft, together with some of the results obtained in flight at the Royal Aircraft Establishment.

The new dynamometer serves to measure the torque transmitted from the engine to the airscrew and consists of a suitable spring element inserted in the drive. This spring has a linear torque/deflection characteristic over the working range and its deflection under the transmitted torque is measured by electrical means. The electrical system used has been developed at the Royal Aircraft Establishment from the basic principle of Ford's torsiometer which has been used largely in marine practice.

In essence it consists of a set of electrical transformers whose primaries are energized by a suitable A.C. generator and the variation of air-gaps in the iron cores is used to provide a variation of secondary output voltage which is arranged to be proportional to the change of air-gap. The deflection of the spring element in the dynamometer is arranged to vary the air-gap a like amount and therefore the output voltage is a measure of the torque. The well-known null reading method is employed whereby the output from the dynamometer is balanced by an equal and opposite output from a similar set of transformers, the air-gaps of which are varied by a hand micrometer mechanism. When the outputs are in electrical balance the reading on the scale of the hand micrometer mechanism is a measure of the mean deflection of the spring and therefore of the mean torque. The fluctuations of torque about the mean which are present, due to transient torque effects and torsional oscillations in the shaft system, produce a modulation of the secondary output, the mean value of which is balanced by the hand-operated unit. This modulation being harmonic about the mean is averaged out in its effect on the suitably damped moving coil current instrument which serves to indicate the balanced condition. The system is arranged to be directional in its indications of balance so that positive or negative torques can be measured.

The apparatus has been developed at the Royal Aircraft Establishment and one torquemeter has been in flight use on an engine

power investigation, while another was used in the 24-foot wind tunnel for which it was specially designed as a piece of research equipment.

Both dynamometers have been calibrated under static torque conditions (for which purpose a special torque loading apparatus was designed having particularly small friction and permitting positive or negative torques to be applied without shock by simply rolling a suitable weight along a calibrated beam), and also on the test-bed against a Froude absorption brake when driven by a modern aero engine.

Before accepting these calibrations, which in the first model were accurate within ± 2 per cent., and in the wind tunnel unit within 1 per cent. agreement with the readings of the Froude brake, the dynamometers were run on engine installations driving air-screws and oscillographic records taken of the torque curves, together with the mean torque absorbed by the airscrew at different speeds. This served to check whether the torque oscillations in the drive with the dynamometer fitted were sufficient to cause the limit stops provided across the spring element to touch. No sign of such amplitudes were recorded, although on the test-bed the dynamic system was such as to indicate on the dynamometer readings that the stops were being engaged at one particular speed of running. The range of torque permitted by the stops on the larger dynamometer is from approximately $-12{,}000$ lb. inches to $+60{,}000$ lb. inches, the mean torques of engines for which it is suitable being up to about 50,000 lb. inches, provided there are not any resonant effects greater than $\pm 10{,}000$ lb. inches in torque.

To deal with any circumstances whereby large torque oscillations will occur, a form of construction is proposed for the dynamometer to incorporate a damped system in the drive. In this construction a series spring element is used, part of which is suitably damped in the known manner, and part of which is free, and across which the electrical measuring system is placed. Thus a shaft arrangement is obtained whereby torsional oscillations throughout the system may be adequately damped, while the dynamometer portion is left free to operate under the very much reduced torque variations about the mean value, which is in this way unaffected by the damping introduced. To attempt to place a solid friction damper of the known type directly across the portion of the spring system on which torque is measured, might introduce considerable error in the integration of the torque. To provide a viscous damper having a friction torque proportional to velocity of vibration, which should theoretically leave the mean torque readings unaffected, would prove too heavy, namely, by some 20 to 40 lb., while its truly viscous mode of operation would be always open to doubt.

Reference is made to some typical results obtained in flight on

a radial engine, such as airscrew characteristic curves k_q/J for metal and wooden airscrews, specific consumption curves, and a power curve on the climb at constant engine speed. The paper concludes with reference to some design applications of the dynamometer to modern engines employing variable pitch airscrews.

Radial Engine Rear-cover Assemblies.—The rear-cover units of radial engines contain the drives to the accessories mounted at the rear ends of the covers. Thus, in the example of the Bristol Mercury and Pegasus engines the rear cover has been laid out to provide the most accessible and compact arrangement of the wide range of accessories required for the equipment of modern aircraft. Provision is made for single or dual fuel pump ; high and low pressure air compressors ; shaft-driven electric generator ; combined hand and electric starter ; vacuum pump ; high pressure oil pump ; and constant speed airscrew governor.

Besides the above-mentioned accessories, the rear cover carries the engine oil pump, oil gauge connection, auxiliary oil feed filter, tachometer drive, magnetos and (if required) the control valve for a controllable pitch airscrew.

These sub-assembly units are tested separately with a special test equipment, prior to erection in the engines. The object is to carry out endurance tests on the rear cover and the accessory drives, together with endurance and calibration tests on the oil pump.

The sub-assembly test-unit has an adaptor-plate to carry the rear cover, and through this plate protrudes a tailshaft (identical with that of the engine) and driven by a 20 h.p. variable speed electric motor mounted behind the front panel of the test-stand. Just above the adaptor-plate is the main oil tank which is heated electrically and provided with a thermostat to keep the oil supply to the pump at the correct temperature ; a thermometer is fitted to the front of the tank. The latter is surmounted by a glass oil-measuring container into which the oil delivered from the engine's scavenge pump is passed. This container has an electrical device set to indicate when two gallons of oil have been passed into it. *The oil-measuring device* embodies a warning lamp which is switched on by a make-and-break actuated by the weight of the oil. The container has a hand-operated draining valve which is closed during the taking of a delivery-time reading for the oil pump calibration ; it remains open during an ordinary endurance test, draining the oil back to the main feed tank without allowing the container to fill up.

The delivery from the pressure side of the oil pump lubricates the rear-cover bearings, whilst a pipe-line fitted with an oil control valve takes the place of the main feed to the engine, and is used in conjunction with the relief valve in the pump to obtain the desired pressure and circulation conditions. A gauge on the instrument panel registers the main oil feed pressure.

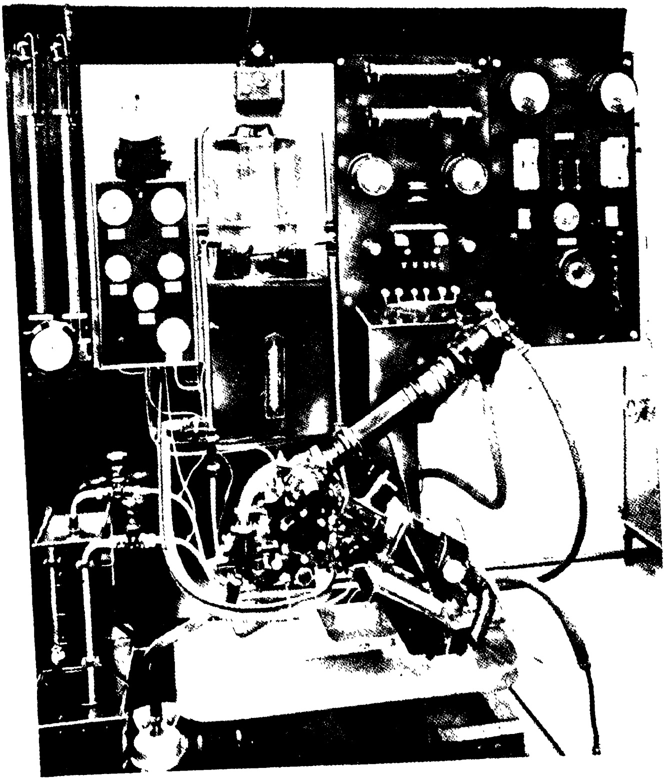

FIG. 341. Testing Bristol radial engine rear-cover units (*Mercury and Pegasus.*)

[*To face page* 388.

The oil drained from the rear-cover bearings, together with that from the main delivery pipe-line, is collected in a lower tank representing the engine sump. From this tank a pipe leads to a filter, and thence to the suction side of the scavenge pump. As the scavenge pump has a greater capacity than that of the pressure pump, the lower tank, like the engine sump, is always dry during running, and the capacity checked by means of the upper calibration container is, of course, that of the pressure pump.

An attachment is included for checking the operation of the high initial oil-pressure device and sprayer valve.

On the left-hand side of each test-unit is a petrol tank, surmounted by a smaller tank fitted with a level-gauge. These are used in conjunction with mercury tube manometers for circulating petrol through the " Bristol " dual fuel pumps on the rear covers under test, so as to load these pumps, and therefore their drives, to represent working conditions. The precise calibration of petrol pumps is, however, carried out on a separate test-rig.

The remaining drives in the rear cover are similarly loaded, either by the use of actual accessories, or by means of electric dynamometers.

In the case of a Mercury or Pegasus type rear cover, as shown in the close-up view (Fig. 341), the electric generator drive is connected by a cardan shaft to an actual generator of the specified type, which operates in conjunction with a loading circuit and instrument panel in the usual way. The R.A.E. low pressure air compressor drive is coupled through a telescopic shaft and suitable gearing to an electric dynamometer behind the panel.

Test Procedure.—The test procedure for a standard Bristol Mercury engine type rear cover is as follows :—

The rear cover is received from the sub-assembly section complete with all its drives, and with all specified external accessories (other than the generator and the low pressure air compressor) fitted in position. It is mounted on the test-unit, and all pipe-lines, generator and air compressor connections, tachometer drive and other details duly fixed. The oil supply cock, oil control cock and petrol valves are opened, and the main driving motor started ; but before accelerating from the lowest speeds, the correct functioning of the oil and fuel pumps is carefully observed. The unit is then gradually accelerated to cruising speed, at which an endurance test of 1¾ hours' duration is carried out. The oil pump capacity is checked by means of the calibration tank, after which the speed is increased to the scheduled maximum for a further quarter of an hour. The oil pump capacity is again checked under these conditions, and the rear-cover assembly is then removed from the test-stand for stripping and inspection. The dismantled parts are thoroughly washed in clean paraffin and visually examined. Minor

running marks may be dressed-up by stoning or burnishing, but if a defect necessitates the replacement of a major component, the entire rear-cover assembly is submitted to a repeat endurance test, after which it is again dismantled as far as may be necessary to examine the condition of the substituted part.

If fully satisfactory, the re-assembled rear cover is in due course built into the engine. After the latter has undergone its standard two-hours' endurance test, however, it is completely dismantled for inspection, which naturally includes the rear-cover parts. The latter, closely examined and once more assembled, are submitted to a further half-hour rear-cover test, followed by oil calibration checks at slow speed, cruising speed and maximum speed respectively, before being accepted for the engine re-build and final test.

Altitude Tests of Engines.—It is well known that the power output of an internal combustion engine, other conditions remaining unaltered, depends upon the density of the atmosphere in which it works. Elementary considerations of the initial pressure at the end of the suction stroke, and its influence upon the compression and mean pressure, will suffice to illustrate this point. The results of laboratory and aircraft tests have shown that for most practical purposes it is sufficiently accurate to assume the horse-power output (at full throttle) of an engine varies as the density of the atmosphere.

Thus
$$H = H_0 \frac{\rho}{\rho_0}$$
where ρ and ρ_0 are the densities and H and H_0 the corresponding power outputs.

A more correct expression, which is frequently employed for aircraft engine design, is as follows :—
$$H = H_0\left(\frac{\rho}{\rho_0}\right)^{1\cdot05 \text{ to } 1\cdot1}$$

In this case the horse-power varies at a rather greater rate than the density.

The following table illustrates the manner in which the power output of an aircraft engine falls off as the altitude increases :—

TABLE XVI

Power Outputs of a 150 H.P. Engine. Compression Ratio 5·3

Density as per cent. of Density at 76·0 cm. Pressure	Barometric Weight cm. of Mercury	Corresponding Altitude in Feet	Average H.P. at 1500 r.p.m. and 0° C.	H.P. as per cent. of H.P. at 76·0 cm. per cent.
100·0	76·0	·0	180·0	100·0
92·1	70·0	2,200	166·0	92·3
81·8	62·1	5,250	140·4	78·0
65·5	49·8	11,500	110·8	61·5
49·5	37·6	19,000	80·4	44·6
33·6	25·6	27,200	50·6	28·1

Temperature Effect on Horse-power.—The results of numerous tests made in this country and by the American Bureau of Standards show that, at constant altitude, or barometric pressure, the horse-power output of an engine falls off progressively as the external air temperature increases. It is necessary, therefore, to correct all horse-power performance results to a standard temperature if the results are to have any comparative values.

The American tests showed that in the case of 150 h.p. and 180 h.p. Hispano Suiza engines (eight-cylinder, water-cooled Vee type) the following were the horse-power temperature relations :—

TABLE XVII

Horse-power—Temperature Results

Altitude in Feet	Temperature of Carburettor Air, °C.	Horse-power. Average
5,300	− 15	141·5
	− 5	138·5
	5	136·5
	15	135·0
10,000	− 20	80·0
	− 10	84·5
	0	82·5
	10	80·5
	20	78·5
30,000	− 20	55·0
	− 10	53·5
	0	52·5
	10	51·5
	20	50·5

The above results show clearly the nature of the variations met with in aircraft practice, and that the horse-power falls off with increasing air temperature practically according to a linear law.

The results may be expressed analytically as follows : If H_o is the horse-power at temperature t_o° C., and H that at t° C., then

$$H = H_o \cdot \frac{529 + t_o}{529 + t}.$$

If the temperatures are expressed in Fahrenheit degrees, viz. T_o° and T° F., then

$$H = H_o \cdot \frac{920 + T_o}{920 + T}.$$

These results apply to the case stated between −20° and +50° C. ; it is probable that the correction factor for other designs of engine and carburettor will be somewhat different.

From theoretical assumptions, however, it can be shown that the horse-power output varies inversely as the absolute temperature.

This leads to a correction factor of about twice that found by experiment. Thus the constants 529 and 920 in the above formulæ become 273 and 491 respectively.

Effect of Both Temperature and Barometric Pressure.—It has been seen that the power output varies as the air density, which in its turn depends both upon the temperature and the height of the barometer. If the air density is determinable, the horse-power results of tests can be reduced to a standard density value by the aid of the expressions already given.

The standard density chosen may conveniently be that of dry air at 0° C. and 760 mm. of mercury pressure.[1]

To correct the horse-power H [2] obtained at any known temperature $t°$ C. and pressure P, to any other temperature $t_o°$ C. and pressure P_o, the following relation may be employed :—

$$H = H_o \frac{529 + t_o}{529 + t} \times \frac{P}{P_o} \text{ where } H_o \text{ is the horse-power at } t_o° \text{ C.}$$

Experimental Determinations.—From what has been stated, it will be observed that although the relationship between the output of an engine and the temperature and pressure of the atmosphere in which it works can be deduced from theoretical principles, yet the practical results may be different. A good deal depends upon the design of engine, its compression ratio, the carburettor design, cooling water temperature, and other factors.

Laboratory Tests of Altitude Effects.—Since most aircraft engines have to operate at various heights above the ground, a knowledge of their behaviour under different altitude conditions becomes essential. Thus, it is necessary to ascertain the effects of low temperatures and low pressures upon the performance, carburation, and cooling of the engines.

It is not possible to carry out scientific tests of such altitude effects by fitting the engines to aircraft and ascending to high altitudes, since there are several factors to be observed, and the necessary test apparatus cannot be carried conveniently in an aeroplane. It has been found possible, however, to make actual measurements of power in the air, using a hub dynamometer, whilst indicator diagrams and fuel consumption measurements have also been made. These tests are not satisfactory, however, from the point of view of engine design and development. To overcome the difficulties encountered in making actual air tests, the American Bureau of Standards, in 1918, designed and built the first altitude laboratory for determining the performance of aircraft engines under the pre-

[1] The standard density selected, in this country, for aircraft instrument calibration purposes corresponds to a barometric pressure of 760 mm. of mercury at 16° C.; this is equivalent to an air density of 1221 grammes per cubic metre.

[2] See also p. 444.

cise conditions of low temperature and rarefied air which exist at high altitudes. The so-called altitude chamber, where the engine is mounted for test, is a reinforced concrete box capable of withstanding an external pressure of a ton per square foot. Powerful vacuum pumps evacuate the chamber and take away the exhaust gases. A shaft carried through the wall of the chamber connects the engine with a dynamometer for controlling the load and measuring the power developed. Connections are also made through the chamber walls for supplying metered quantities of air, fuel, oil, and water to the engine; for adjusting throttle and spark advance; and for measuring the temperatures and pressures in and about the engine. Thus all operating conditions are controlled and all measurements made without entering the altitude chamber during a test.

A number of engines of various types have been tested in the altitude laboratory in question. In addition to tests of particular engines under standard conditions, much general information has been obtained as to the influence of engine design and operating conditions on performance. The effects of temperature and pressure on engine power and the gain obtainable by supercharging are illustrated by results of tests made in the altitude laboratory on a modern 12-cylinder aircraft engines.

In connection, also, with superchargers, the altitude laboratory enables the design and development of new types to be proceeded with expeditiously in comparison with actual aircraft tests at different altitudes.

The latter tests are, of course, essential, but only in the final stages of development when the designs are ready for production.

A brief account of the American Bureau of Standards plant will be given, in order to illustrate the methods adopted to reproduce the atmospheric conditions at various altitudes. The altitude laboratory in question consists of a concrete chamber within which the test engine is mounted, and from which the air may be exhausted to any pressure as low as one-third of an atmosphere by means of a centrifugal exhauster. It is thus possible to simulate conditions at altitudes as high as 30,000 feet and even above. The air can be cooled by passing it over refrigerating coils to a temperature corresponding to the altitude of the test. In the interior of the chamber electrically driven fans are mounted; these circulate the air over the coils and about the engine. The power of the engine is absorbed and measured by an electric dynamometer, and a water-brake mounted outside the chamber and connected to the engine through a flexible coupling.

The original altitude laboratory [1] measured 6 feet 2 inches wide

[1] The altitude chamber was unfortunately wrecked in the autumn of 1923 by an explosion caused by petrol leakage; several of the assistants were killed or injured.

by 15 feet long by 6 feet 6 inches high internally. The walls of the chamber were 1 foot thick, heavily reinforced with $\frac{3}{4}$-inch steel bars to withstand the pressure of the atmosphere outside. There were two doors opening on opposite sides of the chamber, 4 feet by 6 feet 6 inches in size, swinging on hinges and closing against heavy rubber gaskets; the doors were made of oak beams with $\frac{1}{2}$-inch softwood loosely held with headless nails, the whole being covered with airproof paper. Each door was provided with three windows for viewing the engine. The interior of the chamber was lined with cork for insulation, and to guard against excessive air leaks the outside was covered with a very heavy coating of asphalt paint.[1]

The chamber, which is shown in outline in Fig. 342, was divided into two parts, the first containing the engine and the second the cooling coils. The engine was mounted on a special stand. This support was designed for the purpose of duplicating as nearly as possible the flexibility and inertia of the typical fuselage mounting. This design made possible an accurate adjustment of stiffness as regards transverse and vertical vibration and rotation about each of the three principal axes of the engine.

Two oak beams, in this case 2 by 6 inches by 6 feet 3 inches long, were supported at the ends to form the basis of the mounting. The engine was mounted directly on two supplementary beams, of 2 by 4-inch section and of the length required for the particular engine under test. These supplementary beams were free from the main beams except at two points where they were bolted together through a thin separating block. Two yokes were provided to prevent torsion of the individual beams, but had no other effect, as they were free from contact with any other part of the structure.

The Air-cooling Systems.—The air-cooling system may be divided into three parts, the refrigerating plant, the cooling system for the carburettor air, and the cooling system for the interior of the altitude chamber.

The refrigerating plant was installed in the left-hand portion of the building, as seen in Fig. 342. The ammonia compressor was a 9 by 9 inch double cylinder, vertical, enclosed machine, with a refrigerating capacity of 25 tons in 24 hours, and was built by the York Manufacturing Company, York, Pa. It was belt-driven from a 50 h.p. electric motor. The plant operated on the direct expansion system, the ammonia condenser being placed against the outside of the west wall of the building, with the ammonia receiver along the north wall, back of the compressor.

The cooling system for the carburettor air consisted of a bank of

[1] **In altitude chamber design it is advisable to provide special outwardly-opening doors, or spring-loaded flaps, to act as pressure releasers in the event of accidental explosions. These should be placed away from positions used by the test assistants.**

ammonia coils mounted on top of the altitude chamber. The coils were made up of 2000 feet of $1\frac{1}{4}$-inch pipe, enclosed in a box and insulated with 4 inches of sawdust. The air was made to pass through this box in a tortuous path, and then led through an insulated pipe provided with a set of electric heating grids and a regulating valve to the test chamber through opening 18. From this inlet it passed through the air meter to the carburettor. In this way warm or cold air could be supplied to the intake as required.

The system for cooling the air within the chamber was made up of a bank of 800 feet of $1\frac{1}{4}$-inch ammonia coils, placed in the left-hand portion of the altitude chamber. Four motor-driven fans were provided to force the air over these coils, while another fan was installed to circulate the air past the engine itself when desired.

By means of the refrigerating plant and cooling system just described it was possible to reduce the temperature of the air admitted to the carburettor and that within the test chamber to a point approximating the temperature at any altitude up to about 30,000 to 40,000 feet, depending upon the size of the engine. Owing to the fact that the temperature could not be controlled readily by means of the refrigerating plant, the air, after cooling and before admission to the carburettor, was passed over a series of electric grids, by means of which the temperature could be raised again and kept at any desired point. The current flowing through these grids was controlled by conveniently placed switches. Some difficulty was experienced due to the condensation of moisture which occasionally causes a " snowstorm " in the air passage to the carburettor. This difficulty is entirely overcome in the new laboratory, through the elimination of leaks into the refrigerating chamber and the use of what may be termed a " settling chamber," through which the air passed after being cooled, and in which the air-flow was so sluggish that the snow was deposited.

The Jacket Circulating Water-cooling System.—The jacket water-cooling system (Fig. 343) was arranged as follows: Above the altitude chamber was placed a cylindrical iron tank connected to the inlet and outlet pipes of the engine's circulating system, and with another pipe from the city mains, while an overflow led to the sewer. A thermostat was placed within the tank, the brass rod of this device controlling a pilot valve which admitted or discharged city water from a bellows, which in turn controlled the main valve on the city supply pipe. If the water temperature in the tank rose above a certain point the expansion of the thermostat rod caused the pilot valve to open, admitting water to the bellows, and thus allowing cold water from the mains to flow into the tank. When the temperature again fell the contraction of the thermostat rod closed the pilot valve and allowed the water to escape slowly

from the bellows. This arrangement enabled the cooling water temperature to be maintained within 2° C.

The Exhaust Cooling System.—The exhaust pipes connected to the engine were water-jacketed, the inner pipe extending down about 3 feet from the exhaust port, while the outer pipe or jacket was continued from the exhaust port to the main exhaust manifold in the form of a flexible tube. In this way the whole connection was flexible. The water from the annular space mixed with the exhaust gases only at a point some distance from the engine. The

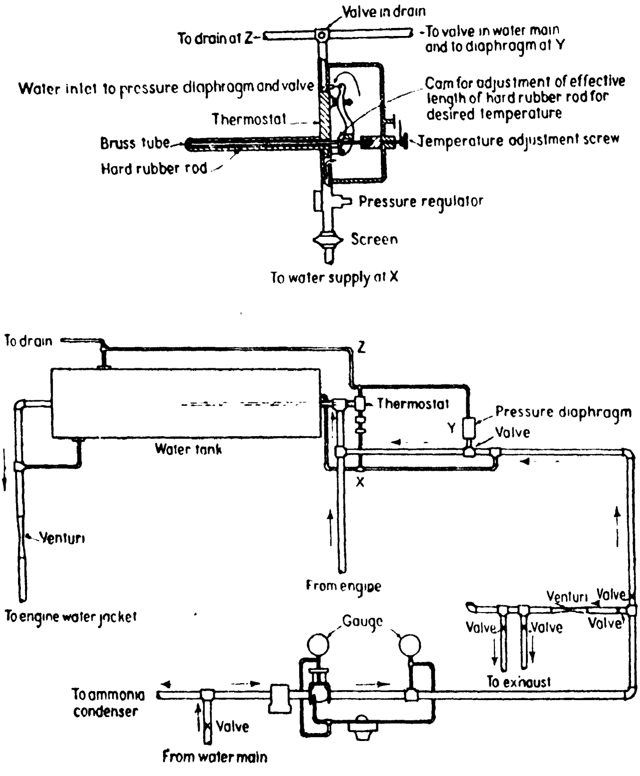

FIG. 343.—Jacket circulating water arrangements.

water entered the altitude chamber through a single opening, and was then distributed to the different exhaust pipes. The mixture of exhaust gases and water passed through two 5-inch pipes to auxiliary exhaust tanks placed just outside the chamber, where the water was drained off whilst the gases passed into the exhauster. The drain pipe had a drop of about 25 feet, with a seal at its lower end, to maintain the vacuum in the exhaust tanks. The auxiliary exhaust tanks were both connected to a 6-inch main, which led to a centrifugal exhauster. Another 3-inch pipe was led from the

main directly to the altitude chamber, and served to withdraw the air from the latter, thus maintaining the barometric pressures on the exhaust and within the chamber approximately equal. By means of a valve communicating with the outside air, placed near the exhauster, the pressures maintained could easily be regulated independently of the speed of the pump.

The exhauster was a Nash "hydroturbine" one with a rated capacity of 1500 cubic feet per minute at a 12-inch vacuum, at 300 r.p.m. It was belt-driven from a 75 h.p. D.C. motor.

General Information.—The electric dynamometer employed was built at the Sprague Works of the G.E.C. The water-brake, previously mentioned, was provided in order to supplement the electric dynamometer. The engine control cables and the copper tubes for the manometers were all carried to two control-boards placed just outside the chamber, and so arranged that one man could control the entire plant whilst being able to observe all of the measuring

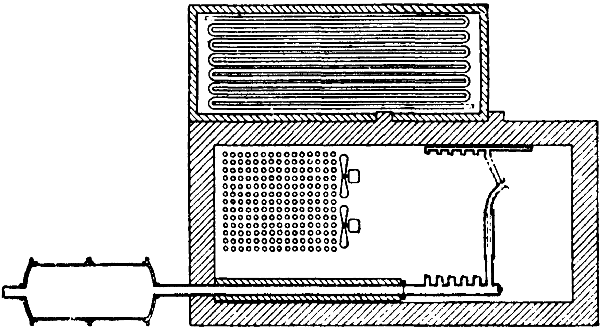

FIG. 344.—The exhaust cooling arrangement.

instruments. The spark and throttle levers worked in graduated quadrants to indicate the exact positions of the levers on the engine.

The instruments provided enabled a considerable number of measurements to be obtained from the engine under test. They included :—

(1) Five Venturi gauges for the carburettor inlet air, jacket water, exhaust cooling water, oil-cooling water, and petrol supply respectively.
(2) Eight manometers for the exhaust back pressure, carburettor, float chamber pressure, average exhaust manifold pressure, carburettor entrance air, Venturi pressure difference, inlet manifold pressure, pressure differences, and individual parts of the carburettor system respectively.
(3) Barometers.
(4) Four vapour thermometers, for the temperatures of the jacket inlet and outlet, and the oil inlet and outlet, respectively.

(5) Oil-pressure gauge.
(6) Revolution counter, provided with magnetic and hand clutches.
(7) Twelve thermo-couples for the temperatures of the oil-cooling water, carburettor air at entrance to the Venturi meter, jacket water and outlet, exhaust cooling water inlet, exhaust water chamber, oil at engine inlet and outlet, carburettor air, inlet manifold and petrol, respectively.

Suitable pipe connections were provided for obtaining samples of the exhaust gases from the engine; the samples were analysed in an Orsat gas analysis apparatus.

Fiat Altitude Testing Laboratory.—The Fiat testing equipment for air- and water-cooled engines under external air pressure conditions equivalent to altitudes up to 32,000 feet is illustrated, diagrammatically, in Fig. 345. A full account of this testing plant was given in *The Automobile Engineer* of July, 1936.

The plant in question embodies: (1) an air-tight test chamber of suitable dimensions, (2) engine test bench, (3) pressure-reduction plant, (4) refrigeration plant, (5) humidification plant, (6) wind-stream plant, and (7) central control-board.

In regard to the air-tight chamber there is an automatic alarm device to give immediate warning should explosive mixture find its way into the free interior of the test chamber.

Refrigeration of the test plant is arranged by means of a brine circulating system instead of ammonia.

The test chamber is fitted with diaphragm pattern pressure relief valves to operate at 1½ atmosphere pressure, thus minimizing the risk of injury to the plant in the event of an internal explosion.

All of the controls and indicators are brought together into a single central control-board; the operation of all controls for decreasing pressure in proportion to altitude is effected automatically at all points.

The engine test plant is divided into three sections, namely, (1) a front one provided with a removable dished head, 39, and track, 40, for placing the engine to be tested in position; (2) an intermediate section A, forming the engine test-cell and a rear section, through which the shaft, 41, passes to connect up with the engine dynamometer, L, outside the test chamber. A manhole, 45, is provided for enabling a person to enter the test chamber. Two thin insulating bulkheads, 43, hinged and detachable, can be used to close the engine cell and insulate it from the rest of the test chamber. A heating apparatus, 44, comprising an electric fan and heating resistance is provided for the purpose of quickly warming up the engine test-cell. The shaft, 41, has three flexible couplings and three adjustable bearings. The power absorption unit, L, consists of a hydraulic brake, 46, and a coupled electric dynamometer, 47;

the former unit can, however, be run separately for lower power absorption purposes.

The dynamometers are operated from the central control-board K, through relays to the various control points on the equipment.

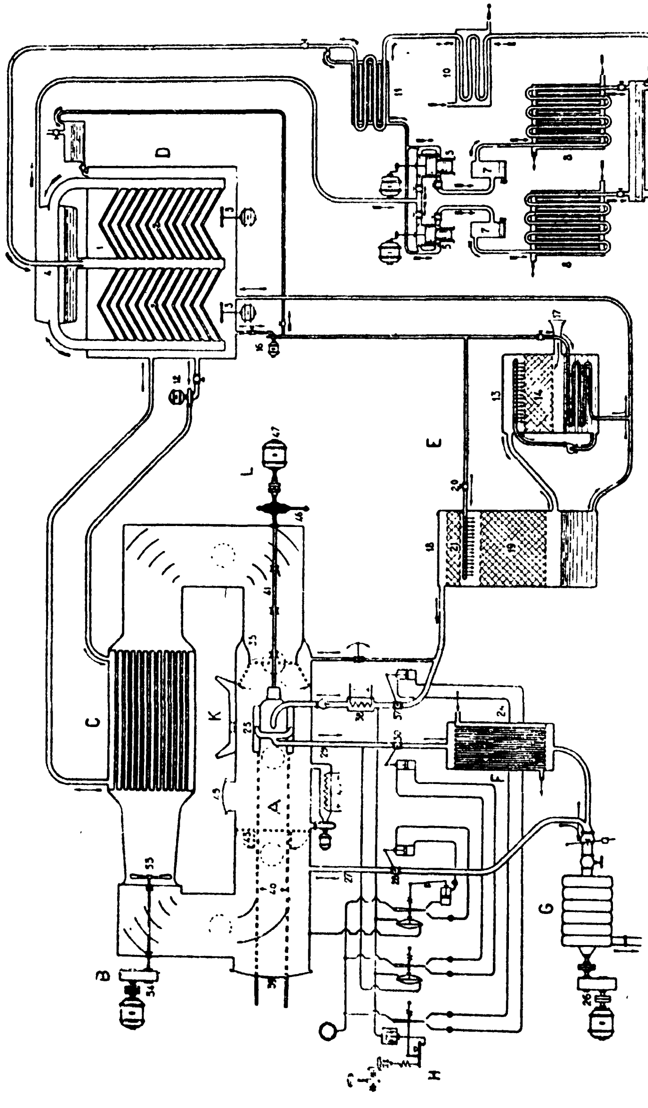

FIG. 345.—The Fiat altitude testing laboratory layout.

The fuel feed system (Fig. 346) consists of (1) a fuel-mixing tank having piping leading from six different tanks, 48, each containing a different grade of fuel, so that the engine can be tested with any of these fuels; (2) a gauze filter, 49, inserted in the fuel

line after the mixing tank; (3) a first float chamber, 50, operating at atmospheric pressure; (4) four fuel meters, 51 and 52, inserted in the fuel piping in multiple, two being of the volume and two of the direct-reading pattern; (5) a second float chamber, 53, having a pressure-equalizing connection with the engine air intake. The float chamber is also provided with a vacuum meter and vacuum adjustment valve for direct admission of outside air; (6) a fuel pump mounted, 57, under the engine test bench; (7) a pressure regulator inserted in multiple in the fuel-feed piping and provided with an overflow return pipe to the regulator.

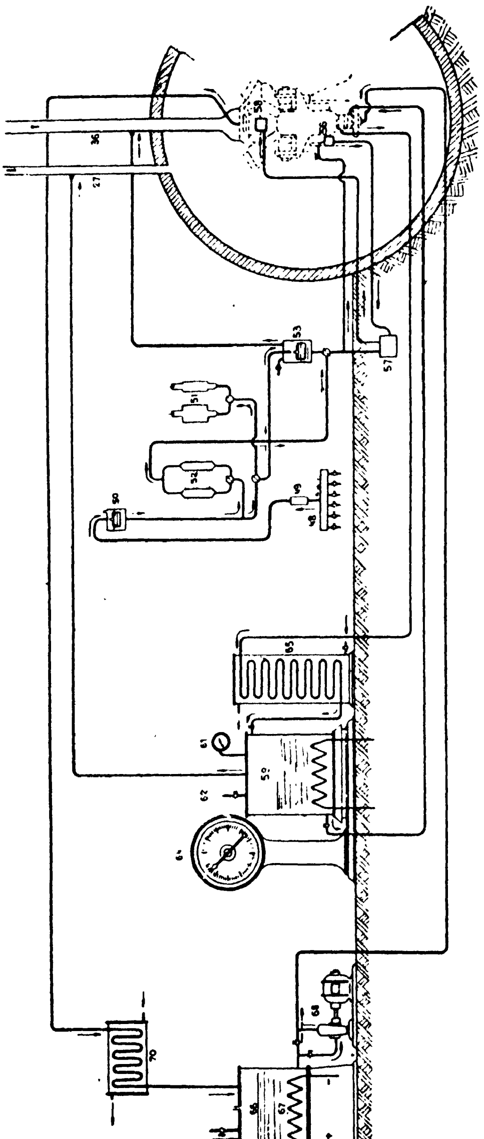

Fig. 346.—Diagram of altitude testing equipment fuel feed, lubricating and water-cooling systems.

The lubricating system consists of an oil tank, 59, connected with the test chamber by a pressure-equalizing pipe, 27, so that it functions at the low pressure therein, and provided with a vacuum gauge, 61, located on the central control-board, with a vacuum adjustment valve, 62, which admits air to the tank directly from the outside, and which is provided also with an electric heating-unit for heating up the oil preparatory to starting the engine. The entire oil tank is mounted on a weighing machine having its dial, 64, located also on the central control-board. Thus, it is possible to check at any moment the oil consumption of the engine under test.

An oil-cooling coil, 65, located in the return line to the oil tank, and consisting of a coil immersed in a counter-flowing current of fresh water. The temperature of the oil entering and leaving the engine is shown by two thermometers with distant reading dials on the outside control-board.

The engine cooling system consists of the following units:—

A cooling water-tank, 66, with a breather, allowing operation at atmospheric pressure, and provided with the usual type water-level glass, and an electric heating-coil, 67, for heating up the water for starting the engine.

A motor-driven pump, 68, for circulating the heated water preparatory to starting the engine. After the engine is started, this pump is cut out and the circulation of water is handled by the water-circulation pump on the engine.

A water-cooling coil, 70, in the return line from the engine, consisting of a coil submerged in a counter-current of fresh water. Thermometers registering water temperatures at the engine inlet and outlet are mounted on the central control-board.

The engine control-board is fitted with the usual temperature, pressure and speed indicators and recorders. In addition there is a hygrometer in the engine intake-air duct, an exhaust-gas analyser, flowmeters, consumption meters, and automatic weighing machines for the fuel, lubricant and intake air. Further, there is a hand-wheel control for the automatic altitude regulator, and a "rapid descent" control.

The air-pressure reduction plant consists of a seven-stage turbo-blower, G (Fig. 345), run at 750 r.p.m. by a three-phase induction-motor, 26, of 300 h.p. through a gear-train which steps up the velocity of the blower to 9000 r.p.m.

The test chamber pressure-reduction piping leading to the turbo-blower in which there is an automatically operated control-valve, 28, which in turn is controlled by an automatic altitude regulator, H. This arrangement makes it possible to obtain pressures in the test chamber as low as a value of 0·26 for the ratio of the absolute pressure obtained divided by the absolute normal atmospheric pressure, which corresponds to an altitude of approximately 32,700 feet. The altitude regulator is shown in the lower left-hand portion of Fig. 345.

The exhaust-gas-cooler piping, 29, leading from the exhaust-gas cooler to the turbo-blower, in which there is also an automatically operated pressure-control valve, 30, operated by the same type of automatic altitude regulator as that used in the test chamber reduction-piping, thus making it possible to maintain the same pressure in the engine exhaust-piping as that maintained in the test chamber.

These two sets of piping are connected to the turbo-blower

through a single manifold in which a check-valve is placed to protect the blower in case of a sharp increase of pressure, such as would occur, for example, in the case of an explosion inside the test chamber. The blower discharges outside the test-room.

The artificial wind plant, shown at B in Fig. 345, consists of a propeller type fan, 33, and driving motor, 34, mounted outside the test chamber below the floor level. It is a 300 h.p. three-phase induction motor running at 975 r.p.m. and drives the fan at 2400 r.p.m. through gears. A wind velocity of 100 m.p.h., corresponding with a movement of 2650 cu. ft. of air per sec. at a pressure equivalent to an altitude of 16,300 feet can be obtained with this plant.

The air humidifying plant for the intake air, 37, is placed between the second air-cooling unit, 17, and the engine, and includes a pressure-reducing valve, 37, operated by the same automatic altitude regulator, H, that has already been mentioned. This valve automatically reduces the pressure of the intake air to that corresponding to the altitude of the test in progress.

A heating chamber, 38, fitted with electric heating-coils operated by a rheostat located on the central control-board, K. The air is warmed in this chamber up to the temperature corresponding to the altitude of the test under way, and is then led to the air intake on the engine.

Altering the pressure and temperature of the intake air causes a corresponding variation in its humidity and is indicated by the hygrometer on the control-board, K.

The refrigerating plant comprises :—

A refrigerating machine ; located at D.

The cooling surfaces which absorb heat from the air enclosed in the test chamber ; located at C.

The cooling surfaces which absorb heat from the air fed to the carburettor of the engine ; located at E.

The cooling surfaces which absorb heat from the exhaust gases flowing from the engine ; located at F.

The refrigerating machine is of the ammonia-absorption type, using a circulation of a calcium-chloride-brine solution of 32° Baumé density and a freezing-point of − 67° F., which, when the test-plant is in operation, is kept constantly circulating over the cooling surfaces.

For a fully detailed account of the refrigerating plant the reader should consult the original article mentioned previously.

CHAPTER XIII

MISCELLANEOUS INSTRUMENTS AND METHODS

Measurement of Engine Speed.—The crankshaft speed, in revolutions per minute, is measured with an instrument known as a *tachometer;* in some instances the same instrument includes a total revolution counter operating upon the cyclometer or electric contact principle.

The tachometer is generally arranged in two principal forms, namely, (1) for direct reading and attachment to the engine, and (2) for distant reading purposes. There are several alternative types of tachometer used for engine speed measurements.[1] These include: (1) *Mechanical types* driven from and at the crankshaft, or at a distance, by means of a flexible cable connection; (2) *Electrical types* such as the generator and voltmeter ones for alternating or direct current, the solenoid-operated chronometric and the commutator-condenser instruments; (3) *Pneumatic types:* these consist, essentially, of an air pump and a pressure gauge. The pump is attached directly to the tachometer and develops a pressure depending upon the speed of the engine. It is connected by means of copper tubing with a pressure gauge, the dial of which is graduated to read r.p.m.

Mechanical Type Tachometers.—These include the centrifugal and the chronometric types.

The principle of the former instrument is shown in Fig. 347. The centrifugal force produced by the rotation of weights is balanced by a spring. The deflection of this spring is a measure of the speed of rotation and is indicated by a pointer after magnification by means of a suitable mechanism.

The centrifugal element is similar to that of the fly-ball governor and usually consists of two or three brass weights A, each pinned to 2 links L. The upper links are attached to sleeve D which is clamped to shaft S, and the lower links to sleeve E, which is free to slide along the shaft. The two sleeves are held apart by the helical spring B. The flexible drive shaft is connected to shaft R and drives shaft S through gear G. As the speed of rotation of the weights is increased, they fly outward and draw sleeve E upwards, thus compressing spring B until the centrifugal force is balanced by the force exerted by the spring. A pin or shoe F held in bearing on the sleeve E by the hair-spring H is deflected upward as the spring is compressed. This

[1] A good account of the various types of tachometer is given in "Aircraft Power-Plant Instruments," Harcourt Sontag and W. G. Brombacher, N.A.C.A. Report No. 466.

deflection is amplified and transmitted to the pointer through the sector and pinion as shown in Fig. 347.

The chief difference between the various makes of tachometer lies in the method of transferring the deflection of the sliding sleeve to the multiplying mechanism ; practically all of the wear affecting the calibration of the instrument occurs here. The centrifugal type of tachometer is relatively simple in design and will withstand hard usage. It has no time-lag—as with certain magnetic and electrical types of tachometer—and therefore gives practically instantaneous speed readings. It will also operate equally well in either direction of rotation of its shaft. The mechanism is readily adjustable to correct for errors of indication, but in general the low speed readings are not quite reliable. It is liable to errors caused by wear after appreciable service and is not easy to lubricate satisfactorily. The frictional drag of the flexible driving shaft increases greatly at low temperatures.

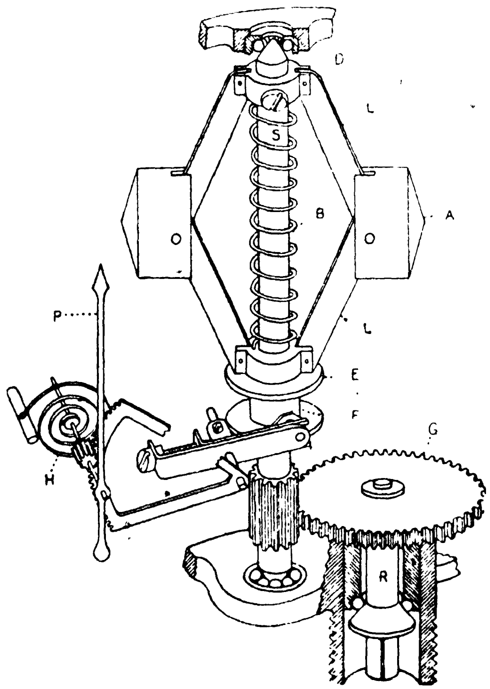

FIG. 347.—Principle of centrifugal type tachometer.

The illustration (Fig. 348) shows the Jaquet[1] hand tachograph, a centrifugal tachometer having also a recording device. A square-centre is provided on the right-hand side for pressing against the shaft centre. This instrument can be used for speeds of 50 to 24,000 r.p.m., and allows records of wide speed fluctuations, such as the acceleration, or the running down speeds of engines ; it is also useful for checking the speed control of engine governors. The record strip is fed forward between two clockwork-driven feed rolls ; the clockwork is wound up by the thumb nut A, whilst the triggers G (at the back, and not visible in Fig. 348) and H and F act respectively to stop or vary the speed rate. Narrow discs are mounted on the top feed roll at positions corresponding to the calibrated revolution rates indicated

[1] Supplied by Messrs. Markt & Co., Clerkenwell Road, London, E.C.

on the dial of the tachometer; these discs are in contact with an inking roll C, and mark on the record strip parallel lines, to give an ordinate scale. The revolution rate during test is read off the dial. There is also a time recorder J, which marks the record at definite intervals of 1 or $\frac{1}{5}$ second as desired.

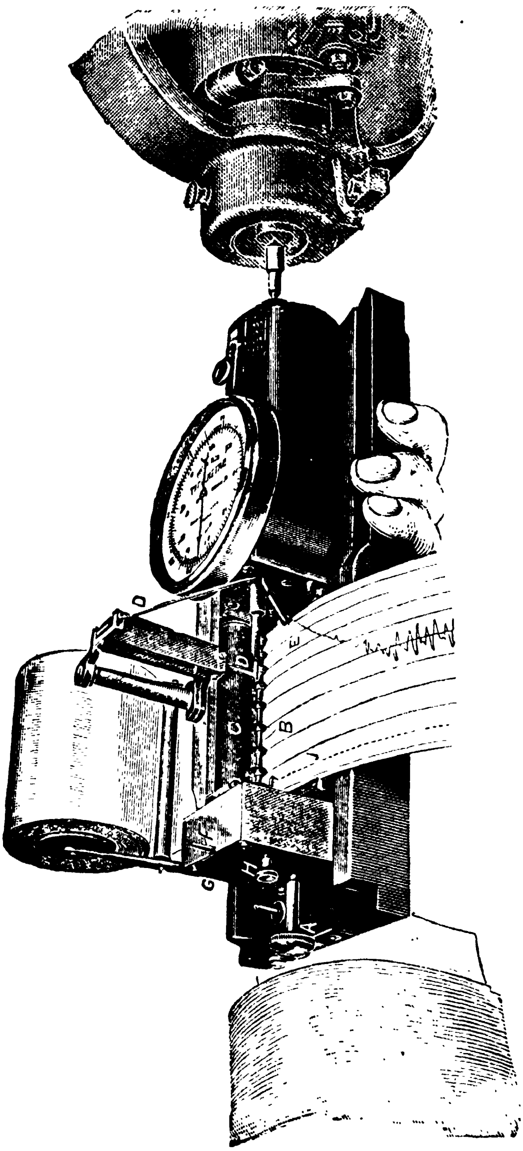

FIG. 348.—The Jaquet hand tachograph.

The Chronometric Tachometer.—This instrument is essentially a revolution counting device, the operation of which is governed, automatically, by an escapement mechanism so as to periodically integrate and indicate the number of revolutions of the drive shaft occurring during each cycle of operations. Its essential components include: (1) the driving mechanism; (2) the escapement mechanism; (3) the power supply for the latter; and (4) the counting mechanism.

A typical tachometer of this kind is shown in Fig. 349. The driving mechanism is indicated at A and B. It includes a mechanism to rectify the motion of the drive shaft into one given direction. The escapement mechanism, shown at C, governs the speed of rotation of the cams, one of which is shown at J. The power supply for the escapement mechanism is contained in the drum M, and comprises a spiral spring, the

inner coil of which is fixed to the shaft connected with the driving mechanism, while the outer coil normally bears with some friction against the inner cylindrical surface of the drum. This mechanism transmits a torque sufficient for rotating the cams, slippage occurring when the torque becomes excessive. The counting mechanism is identified by gears D and H. Its function is to produce a deflection of the pointer corresponding to the speed of the driving mechanism.

The operation of the instrument may be understood by again referring to Fig. 349. A counting gear D is first placed in mesh for a period of one second with gear F which is actuated by the drive shaft. The engagement of the two gears D and F is produced through the intermediate gear E actuated by a cam, one lobe of

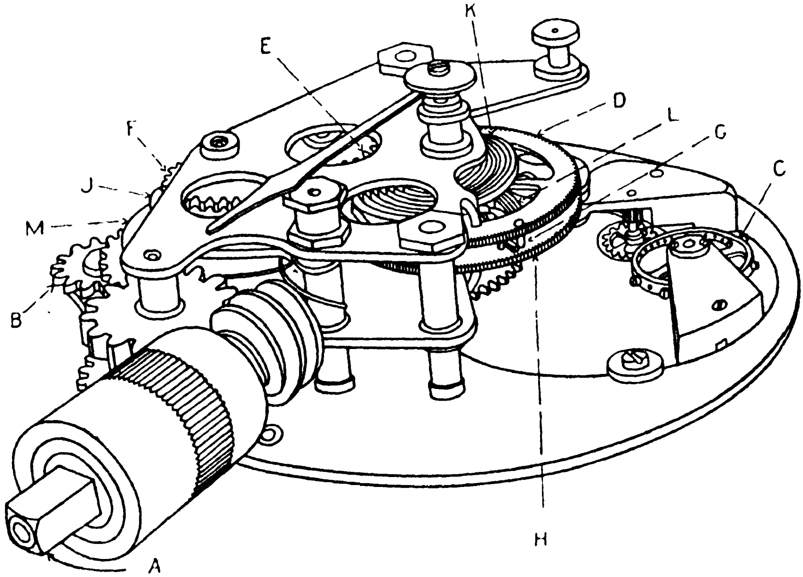

FIG. 349.—Chronometric tachometer.

which is shown at J. By means of a pin and floating link mechanism shown at G, gear H and the pointer, which is rigidly fastened to the same shaft as gear H, are caused to rotate through an angle proportional to the total number of revolutions over a period of one second. Gear H is essentially a ratchet gear and is provided with a pawl which is also actuated by a cam (not shown in Fig. 349). At the end of the one-second period the pointer remains stationary while the counting gear D is disengaged, and, by means of the hair-spring K, returned to its initial position. At the end of the following second the cycle is repeated. If the speed of the drive shaft has increased, the pointer is caused to increase its reading to correspond with the new average speed. If the speed has decreased, the pointer is released by the aforementioned pawl and under the influence of

hair-spring L, returned to a position corresponding to the decreased speed.

The complete cycle of operation requires two seconds of time. When the speed varies greatly, the resulting periodic fluctuation of the pointer is disconcerting. When the speed varies only slightly, the pointer will change its position at the end of every two-second interval by steps of 10 r.p.m., due to the fact that the number of teeth on gear H is 250 while one revolution of the pointer corresponds to a speed range of 0 to 2500 r.p.m.

Other commercial tachometers of this class include the Jaeger, Haslar, and Bonniksen ones, with ranges up to 3500 r.p.m., and cycle periods of 1 to 2 seconds.

Notes on Chronometric Tachometers.[1]—In comparison with the centrifugal tachometer three favourable characteristics are outstanding: (*a*) Low speeds in the range 0 to 500 r.p.m. are indicated; (*b*) the indications have an equal or greater initial accuracy and maintain this accuracy throughout the life of the instrument; and (*c*) the indications are free from lag due to friction in the mechanism. In addition to these comparative advantages the chronometric instrument has (*d*) a scale uniformly divided in units of r.p.m. and is (*e*) easy to adjust for minor deviations from the proper calibration which is done by adjusting the period of the balance wheel.

The instrument suffers by comparison with the centrifugal tachometer in that (*a*) the indication follows changes in speed at intervals of 1 or 2 seconds, depending upon the design, which experience shows is troublesome in estimating the average speed during minor fluctuations, and (*b*) the average speed of the previous interval of time is indicated, not as is more desirable, that at the instant of observation. The centrifugal tachometer indicates the instantaneous speed with a lag caused by the inertia of the mechanism, which has, however, the practical advantage of smoothing out the minor rapid fluctuations in speed, thus aiding the observer in determining the average speed.

Accurate Speed Measurements.—For very accurate measurement of the average speeds it is necessary to employ a revolution counter and stop-watch. The readings of the counter is taken at the commencement and finish of the test, and the time of the test by means of a stop-watch; the difference in the counter readings, divided by the time interval, gives the average rate of revolution of the engine crankshaft. Most revolution counter devices are provided with clutches for engaging with the shaft, the speed of which is to be measured. The usual method of clutch engagement is that of pressing the counter driving-shaft end (usually provided

[1] *Ibid.* page 403.

with a squared conical driving centre) into a centre drilled in the end of the engine shaft.

It is generally possible to arrange a simple lever or cam device which will allow both the counter to be engaged and the stop-watch to be started with one movement of a lever, and at the finish of the test the counter to be disengaged and the watch stopped by another movement of the lever.

The more common types of revolution counter work on the multiple train of wheel principle, similar to the Veeder counter and well-known bicycle cyclometer.

Electric Tachometers.—A more convenient form, and one which enables the counter dial to be placed in any desired position remote from the engine, works on the electric principle. One type consists of a simple wipe-contact, or make-and-break, placed on the engine shaft, and so arranged that once every revolution contact is made, and when the circuit switch is engaged, the circuit is completed. The electric current from a two-volt cell or accumulator in this circuit energizes an electromagnet in the counter dial. The magnet attracts a pivoted soft iron armature, and the movement of the latter moves a ratchet wheel through a tooth. The dial of the counter unit is graduated in units, tens, hundreds, and thousands, a suitable counter gearing being provided for this purpose.

In this way the recording unit can be placed on an instrument board near the operator, and under his control. It is necessary, merely, to close and open the switch to start and to stop the counter. Another somewhat similar system is that described on page 97, and illustrated in Fig. 102, page 143.

The Record[1] electric tachometer (Fig. 350) consists of two units, namely, a generator and a voltmeter.

The generator is driven off the engine crankshaft and consists of a permanent magnet current generator having cobalt magnets in place of the usual electrically excited field coils. It is therefore entirely self-contained and requires no outside source of electrical supply.

It is made in both the A.C. and D.C. patterns. The former is somewhat cheaper but the readings are uni-directional, i.e. a positive reading of the voltmeter is given, irrespective of the direction of rotation of the generator armature. The D.C. type gives both forward and reverse readings, according to its direction of rotation and is therefore suitable for reversible engines. The generator has a drum-wound armature and runs on ball-bearings, the D.C. type generating current at 100 volts for a speed of 2000 r.p.m. (for full-scale deflection); speeds up to 5000 r.p.m. are obtainable with A.C. generators.

[1] Record Electrical Co., Ltd., Broadheath, Manchester.

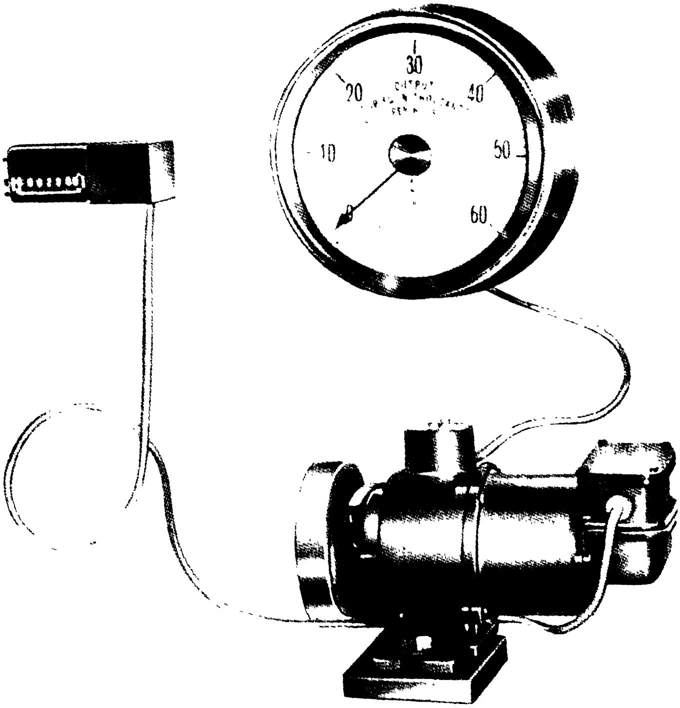

FIG. 350 The Record electric tachometer with contact maker for operating distant electric counting train in conjunction with r.p.m. indicator.

[*To face page* 408.

The indicating voltmeter has a scale graduated in r.p.m., the voltages being proportional to the generator speeds. Various models of the tachometer are made. Another " Cirscale " hand tachometer is made on the same principle for making quick speed readings of shafts by pressing the conical end of the generator shaft into a countersunk centre on the engine shaft.

Stroboscope Speed Indicators.—The speed or frequency of a rotating or oscillating part can be measured by simply viewing the part with one of the hand or stand stroboscopes supplied for the purpose. The speed of the stroboscope disc is regulated until a mark on the part observed appears to remain stationary, when the speed of the stroboscope is read off. The S. G. Brown and Whidbourne direct-reading hand-operated stroboscopes are easy to use, whilst the Drysdale and Crompton types require electric motors for their operation. The principle of the method will be apparent from the account which follows.

Hand Stroboscopes.—The portable type of stroboscope is undoubtedly more convenient for engine and instrument research work, as it can be taken to the object whose speed of rotation is to be measured. Moreover, the speed can be measured at a distance without having to make any mechanical connections with the rotating object.

Hitherto, the only available measuring types of stroboscope have been both bulky and heavy, requiring the use of spring motors, gear trains, automatic speed regulators, electric motors, etc.

The Elverson Oscilloscope is an instrument of its type. It has a slotted drum rotated by means of gear trains (to give different speed ranges) from a clockwork motor provided with a governor.

It is possible with this instrument to measure speeds of rotation or vibration frequencies, and also to obtain slow-motion views of a rapidly rotating body.

The Whidbourne stroboscope, originally invented for Air Ministry purposes, illustrated in Figs. 351 to 354, employs an ingenious air turbine of simple construction, actuated by a rubber bulb through a short length of rubber tubing. It is possible to speed up the slotted drum, through which the moving object is viewed, very quickly, and to maintain it at any speed with ease, merely by regulating the " puffs " given by the rubber bulb.

It is possible in the " visual type " instrument to spin the slotted drum up to about 8000 r.p.m. in a short period of time. The " recording-type " stroboscope has a larger and heavier drum that can be spun up to 4000/5000 r.p.m. in a few seconds.

The *visual type instrument* is illustrated in Fig. 351. It is provided with a hooded viewer to cut off extraneous light, and there is a non-return type of air valve between the rubber bulb and the jet to give a more continuous air-flow.

This instrument is particularly suited to the observation of rapidly rotating or vibrating bodies, such as ordinary machinery parts, springs in operation (e.g. petrol engine valve springs), spinning and other textile machines, electrical machinery, etc. By varying the speed of the drum so that it is slightly less than that of the observed object, the latter will appear to give a slow-motion

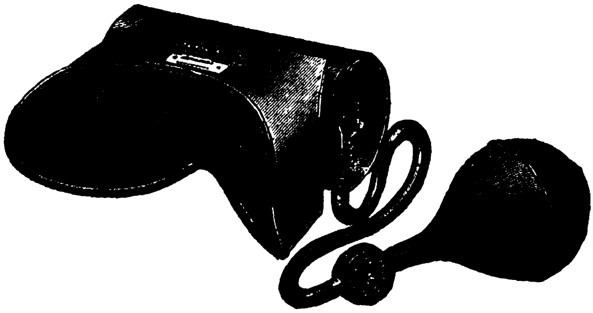

FIG. 351.—The Whidbourne stroboscope.

picture of its cycle, thus revealing any faults too rapid for the eye to observe in the ordinary way.

The *recording pattern instrument* has a modified Haslar rate of revolution recorder fitted so that the speeds of rotation (of vibration) can be measured accurately.

The instrument in question is provided with a clever optical device whereby its ordinary speed measurement range of about

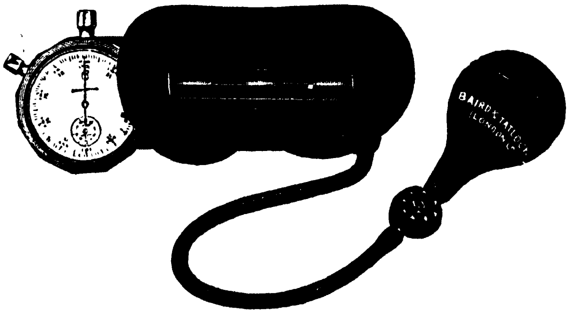

FIG. 352.—The recording pattern Whidbourne stroboscope.

150 to 4000 r.p.m. can be extended ten times, i.e. 1500 to 40,000 r.p.m.

The Drum and Viewing Arrangement.—In order to allow the object to be viewed with both eyes, a pair of slots is provided along the drum, namely, one for each eye ; the hooded viewer is accordingly designed to enclose both eyes. To permit the observation and measurement of speeds much higher than the maximum rational speed of the drum itself, a series of ten pairs of slots

arranged in the form of intersecting chords of the drum (in cross-section) is provided. The eye therefore sees ten times the number of images for one revolution of the drum, so that speeds up to ten times the maximum attainable speed of the drum can be measured.

Fig. 353 shows a cross-sectional view of the drum, and also the viewing arrangement for normal speeds. In this case there is a

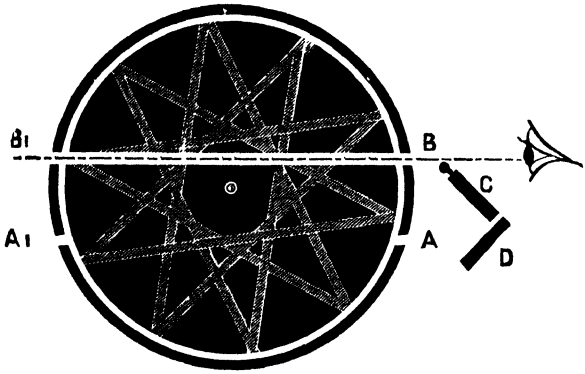

FIG. 353.—Direct normal speed viewing position.

single slot BB_1 having no relation to the other slots (shown shaded), and the eye simply observes through corresponding holes, B and B_1' in the casing of the instrument.

When it is desired to observe higher speeds the hinged mirror C —which extends along the length of the drum—is swung over into the position shown in Fig. 354. The eye then sees the object by a

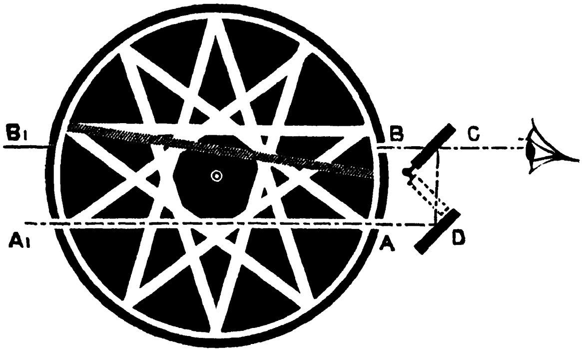

FIG. 354.—High-speed indirect viewing position.

double reflection from the parallel mirrors C and D, and through the slot AA_1. The use of two mirrors gives a double reversal of the image, and there is thus no inversion effect.

It will be observed, from Fig. 354, that each of the ten slots forming chords of the drum circle will come opposite the slot AA in turn, during one revolution of the drum.

When using the arrangement shown in Fig. 352, it is necessary to multiply the Haslar instrument readings by 10 in order to obtain the true speed of the object observed.

Using the Stroboscope.—The Haslar instrument is provided with two knobs, one for starting the recording apparatus, and the other for returning the needle to zero. To make a measurement of speed, if one has a very rough idea of the order of the speed, that is to say, whether it is 500 or 5000 r.p.m., the drum can be rotated by means of the air turbine apparatus described, using the Haslar indicator as a measure of the speed ; thus, if it is known that the probable speed of the object is between 4000 and 5000 r.p.m., one speeds up the drum of the stroboscope, and makes a rough check with the Haslar indicator by pressing the starting knob until one sees that the drum is rotating at a rather higher speed, say about a few hundred r.p.m. above the estimated speed. The object is then sighted with the stroboscope, when, as the speed of the drum diminishes, one gradually sees the rotating object come to rest. The drum is then given a few slight impulses by means of the bulb and air jet, so as to keep the rotating object still whilst the speed is being measured by the Haslar indicator ; the latter measurement takes only a few seconds. As with this form of indicator the needle stops on the actual speed measured, the instrument can be read at any time afterwards.

Accuracy of Readings.—The weight of the complete instrument, as shown in Fig. 352, is less that 2 lb., so that it can be used for very long periods without fatigue. It is compact in design, the drum measuring only about 2¼ inches diameter by 4 inches long. It has a high degree of accuracy, the results of official tests showing that it is possible for ordinary observers to take hundreds of consecutive readings to an accuracy within 0·2 per cent., i.e. 1 part in 500.

Applications.—The instrument described, now available in the commercial form,[1] has been used in various Government establishments for testing engine valve gears, exhaust gases, cams, valve springs, gun-running gear, and propellers. With this instrument air-speed revolution indicators have been calibrated from a distance merely by observing the rotating propellers. A further interesting example of its application is that of the observation of aeroplane propeller speeds whilst the machines were actually flying. An observer standing on the ground, having the stroboscope spun to a slightly higher speed, can quickly bring the propeller blades to rest in the instrument, and thus measure the speed of rotation. High-speed gyroscopes, turbines, rotors, and similar high-speed apparatus can readily be observed and measured with the stroboscope.

[1] Marketed by Messrs. Baird & Tatlock Ltd., London.

Some Other Commercial Stroboscopes.—Among the available commercial stroboscopes mention may be made of the Ashdown "Rotoscope," the French "Stroborama," Zeiss-Ikon and American Westinghouse Electric Company's "Stroboglow"; many other experimental types have also been evolved for special research requirements.

The Ashdown "Rotoscope" operates on the rotating slotted drum principle and permits of binocular visual effects. The drum is driven through a train of gears from a clockwork motor. Each opening of the drum slot is provided with thin opaque blades (Fig. 355) parallel with the cylinder axis, so as to divide the field of vision and thus to produce a shutter action; the sharpness of definition can be increased by increasing the number of blades, but at the expense of loss of brightness and reduced duration of exposure. Another method of increasing sharpness employs the "heteroptic" principle in which a co-axial outer sleeve surrounds the inner slotted drum. The latter rotates at the frequency of the object being observed whilst the outer slotted sleeve rotates at a speed which is a multiple of this frequency—usually at nine times the inner drum speed. The better definition of this type is, however, obtained at the cost of a loss of brightness.

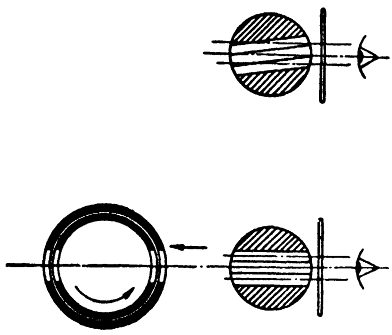

FIG. 355.—Principle of Ashdown rotoscope. (*a*) Normal bladed shutter, (*b*) heteroptic shutter.

The "Stroborama."[1]—This belongs to the flashing light class and employs a special design of neon tube, having no appreciable lag in response. The ordinary neon tube has poor illumination properties and generally requires the use of a darkened room for observations on vibrating or rotating bodies. The Stroborama neon tube, however, takes 1000 watts and produces an illumination of 1000 candle-power.

Fig. 356 shows the priming and lighting circuits and the contact-breaker unit. The latter is driven either by the engine or machine shaft, or by a variable speed motor. The contact-breaker is in circuit with the neon tube and also with a priming current condenser of low capacity. This condenser is kept charged to a few hundred volts by a motor-generator which converts the alternating current of the supply into direct current. The contact-breaker discharges the condenser through the neon tube, this discharge constituting the priming current. The motor of the motor-generator set is in

[1] Described fully in *The Engineer*, Sept. 21, 1934.

parallel across the supply mains with the primary winding of a high-frequency transformer. From the secondary winding of this transformer, current at 15,000 volts, after rectification by a pair of valves, is taken to charge a powerful battery of condensers. These condensers are in circuit with the neon tube and a spark gap. The resistance of this spark gap is adjusted to such a value that when it is added to the resistance of the neon tube the combined amount is too high to permit the battery of condensers to discharge. When, however, the priming current passes through the neon tube the resistance of that tube is reduced and the battery of condensers is enabled to discharge across the spark gap and through the tube. In this way the contact-breaker has to deal only with the low-tension priming current, the high-tension illuminating current being confined to the other circuit. The illuminating current circuit has practically no self-induction, and as a result the illuminating flash is of extremely short duration and of high intensity and follows the movement of the contact-breaker without sensible time-lag.

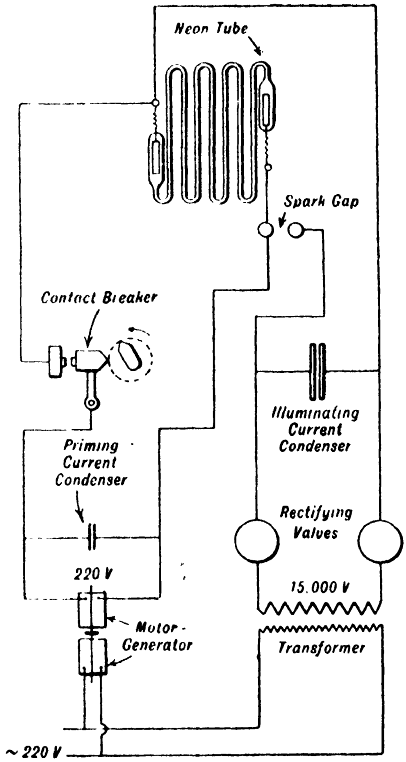

FIG. 356.—The Stroborama priming and electric circuits, with contact-breaker.

A tachometer is provided with the contact-breaker unit for speed indication purposes. With the instrument described, photographs of rapidly moving parts or rapidly occurring phenomena can readily be taken; the duration of the exposure is of the order of a millionth-second, but several superimposed exposures can be made with the special manipulating device provided. In this way, records of rapid movements of parts of engines and machines, aircraft airscrew blades, fuel injections, etc., can be obtained. The Stroborama can also be employed in conjunction with a cinematograph camera to obtain film photographs. In this application, after each exposure of a millionth-second the synchronizer is automatically stepped on through a slight movement so that the film records a slow-motion picture of the phenomenon.[1]

[1] The English agents for the Stroborama are J. Cochrane & Co., 101 Leadenhall Street, London, E.C. 3.

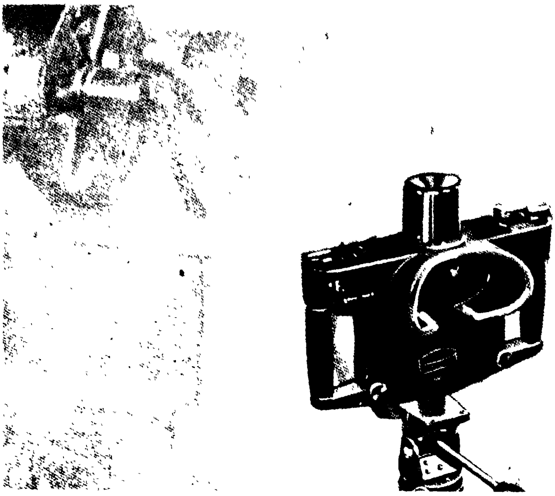

FIG. 358 The Zeiss-Ikon stroboscope mounted on stand for observing moving parts.

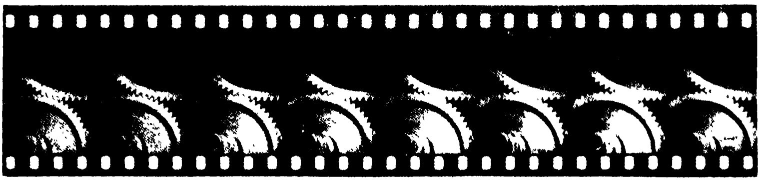

FIG. 359 Stroboscope cinematograph film of running gear-wheels.

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The Zeiss-Ikon Stroboscope is a portable instrument of the binocular vision class and has a single disc mounted on the shaft of a small electric motor, the speed of which is finely adjustable, up to about 2500 r.p.m., by means of a variable resistance and eddy-current brake. Discs with different numbers of slits (1, 2, 3, 6, 10, 12, and 24 slits) are provided and are interchangeable on the motor shaft; those with a small number of slits are intended for the study of comparatively low-frequency motions, and those with a large number of slits for high-frequency motions (up to 1000 cycles per sec.). The slit disc also serves as a flywheel for the small motor. By means of a built-in tachometer, speeds up to 120,000 r.p.m. can be determined, without, of course, touching the object. A valuable feature of this instrument is that a prismatic binocular telescope can be fitted to it which enables one to observe the object at a distance and also gives an enlarged picture. This makes possible the observation of small objects at a distance, as, for example, the rotor of a gyroscope. A tripod stand is provided for the instrument when it is to be used for stationary location purposes.

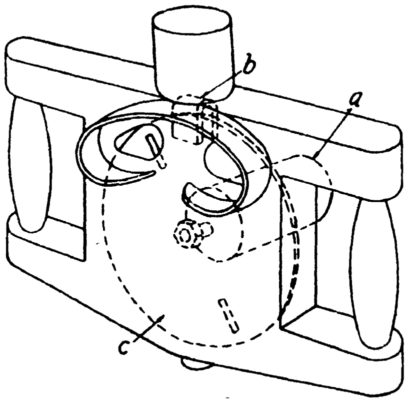


FIG. 357.—The Zeiss-Ikon stroboscope. (*a*) universal motor, (*b*) eddy-current brake, (*c*) slit disc.

The stroboscope can also be used for photographing objects observed. With a single lens camera single pictures can be taken if the object is strongly illuminated, as, for example, with one or more 500-watt Niraphot lamps and reflectors. Cinematograph pictures can also be obtained with the aid of a cine-camera and a " creeper-gear " equivalent device, which in this case utilizes the variable resistance to increase or decrease the speed of the stroboscope disc about the synchronous speed. A typical application is that of the photographing of the meshing of a pair of spur gears rotating at a high speed, so that the results of backlash and incorrect meshing can be studied in detail (Fig. 359). If knocking and oscillation occur, the teeth do not mesh with a regular motion, and one of the two engaging teeth will move up and down—as observed visually or photographically; jerks or oscillations indicate that the gears are not meshing properly. The stroboscope in question has been used for observing the action of high speed metal cutting tools, the movements of spindles of textile machines, petrol engine valves, cams, contact-breakers, etc., electric motors and generators, grinding spindles, typewriter carriage escapements, etc.

The Stroboglow Stroboscope.[1]—This instrument has been designed for studying any periodic movements from 100 to 30,000 cycles per minute and it produces a momentary illumination of 300,000 candle-power. It consists of an electronic timer and lamp unit for producing accurately timed impulses which cause the lamp to flash at any desired frequency. There is also a power unit which is mounted permanently in the case of the instrument, and it contains the electric power supply for the whole equipment ; it operates from a 115-volt, 50-60 cycle A.C. supply (Fig. 360).

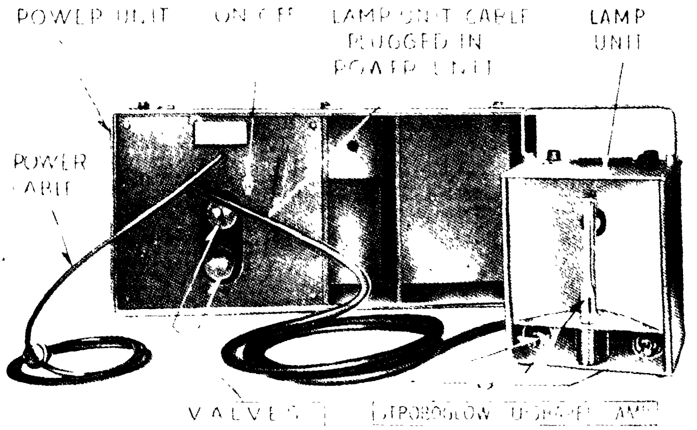
The lamp unit is removable from the case and may be moved around conveniently in actual use. It carries all necessary flashing speed controls thus eliminating the necessity of frequently returning to the main unit to make adjustments. The Stroboglow lamp is mounted in this unit. This " U "-shaped lamp has ends of unequal length to insure correct insertion. Below the reflector are sockets for the amplifier and timing tubes.

The electronic timer consists of a " super-control " amplifier tube, whose grid circuits are controlled by potentiometers and a range knob. This tube periodically charges a condenser which, when it reaches the proper potential, causes the grid of a gas-filled timing tube to permit a discharge to pass through its plate circuit. This circuit includes the primary of a special spark coil, whose secondary lead touches the top of the Stroboscopic lamp ; the latter is continuously supplied with filtered direct current from the power unit, but this is sufficient to cause a flash only when the spark coil is energized. The lamp can be flashed by a suitable external source of high voltage such as an automobile ignition system. This requires only the lamp unit and the power unit, and does not require the contactor. The contactor is an auxiliary piece of equipment which requires both the lamp unit and the power unit for operation. The contactor may be used in some cases instead of the electronic timer. It provides a rapid electrical make-and-break, and is driven mechanically by the machine being viewed. The contactor can be used instead of electronic timing, by plugging it at the jack provided in the lamp unit.

The rubber tip on the contactor shaft is placed in the shaft centre of the rotating apparatus. Phase relations can be changed readily, by simply turning the handle of the contactor, without disengaging the rubber tip from the machine.

The principal field of application is in connection with vibration analysis of rotating and reciprocating apparatus. A secondary field of considerable importance is found in sequence analysis of mechanisms which have motions of a recurrent nature, such as sewing machines, moving-picture cameras and projectors, machine

[1] Westinghouse Electric International Company, London.



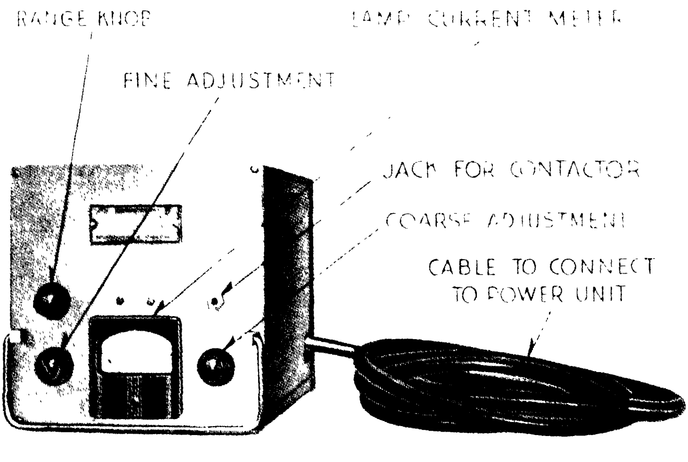


FIG. 360. The Stroboglow stroboscope, showing (above) the power unit and (below) the electronic timer unit.

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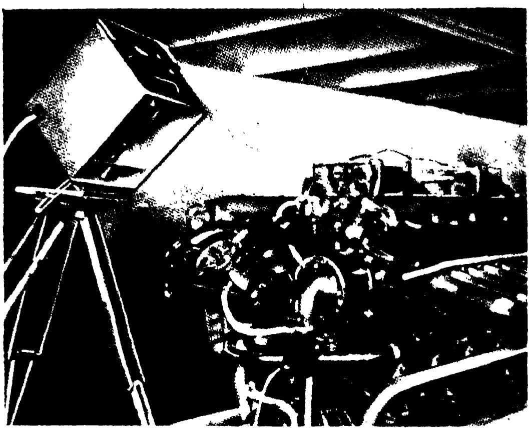

FIG. 361 The Stroboglow stroboscope set up for studying an aircraft engine.

[To face page 417

guns, etc. The Stroboglow may be used for measurement of approximate rotational speeds of machines. With a maximum flashing speed of 5000 per minute, it is possible to analyse motions up to 30,000 per minute, by flashing once every two revolutions for 10,000 ; once every three revolutions for 15,000, etc.

Combustion Phenomena Stroboscope.—A method originated as long ago as 1911, by the late Professor W. Watson, for observing combustion phenomena in, and obtaining spectrographs from, the combustion chamber of a petrol engine employed a stroboscope in the form of a circular disc having a radial slot, the disc

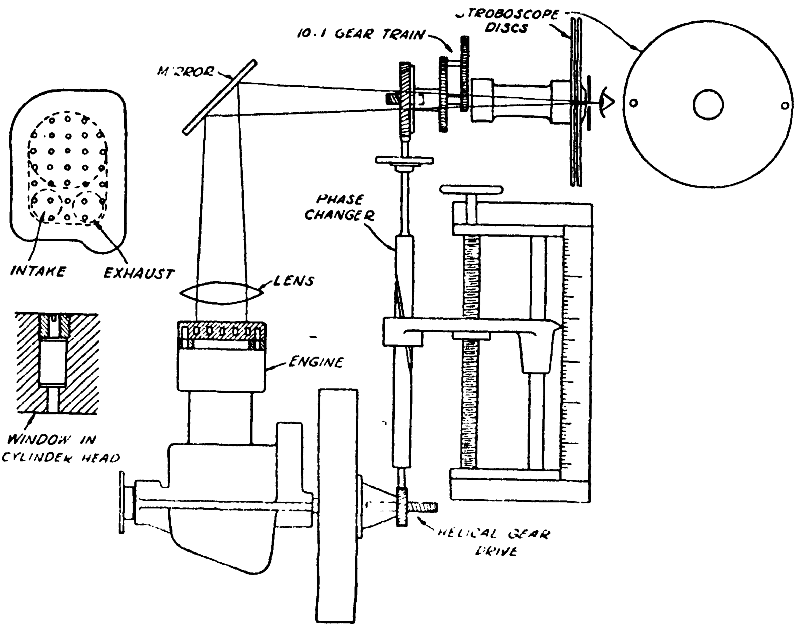

FIG. 362.—Stroboscope for investigation of flame phenomena in internal combustion engines.

being rotated through a differential-type phase changing gear so as to bring different phases of the combustion under observation at will.

An improved method based on the same principle, due to Marvin and Best of the American National Bureau of Standards, illustrated in Fig. 362, employs two co-axial stroboscopic discs, one having ten times the speed of the other ; this refinement enables a considerable reduction to be made in the dimensions of the discs as compared with a single disc. In the type illustrated holes are used for viewing instead of slots. A phase alteration gear, with calibrated scale is employed for obtaining observations at different parts of

the piston's stroke. A number of windows in the cylinder head enables different areas of the combustion to be observed.

Measuring Engine Acceleration.—In some cases it is important to be able to measure the acceleration values of an engine in the interval that elapses after the throttle is opened; this is of special interest in carburettor tests.

Although it is possible to make successive readings of a chronometric tachometer, this method involves the personal element, and cannot usually be considered as sufficiently accurate. If a recording type tachometer, giving engine revolutions on a time base, is available, the acceleration curve can readily be computed from the velocity (or revolutions per minute) ordinates.

The American Bureau of Standards [1] employ a spark accelerometer for this purpose. It consists of a sprocket coupled to the engine through a one-position claw clutch, which draws paper tape, of the same size, and with the same perforation as 16 mm. motion-picture film, between electrodes across which a spark is periodically discharged. This spark is timed by a tuning-fork, which breaks the primary circuit of an automobile ignition coil. The record consists of a series of holes in the tape, the distance between any two holes being proportional to the mean engine speed in that time interval.

In order to facilitate measurement and computation of the results, the tuning-fork was adjusted to give a frequency of 360 cycles per minute. Measurement is made by passing the tape over a sprocket identical with the one attached to the engine. A microscope is trained on the tape, and the sprocket is revolved until a spark hole is under the cross hair. Reading is made on a micrometer head graduation in degrees which is carried on the sprocket shaft. Since the tuning-fork splits a minute into 360 parts and the micrometer head splits a revolution into 360 parts, differences of successive readings give revolutions per minute directly.

Strictly speaking, the data give only displacement and time precisely, and acceleration is obtained only approximately by double numerical differencing. Mathematical analysis shows, however, that the error thus introduced is negligible for most practical purposes. With this apparatus, speed can be measured six times a second, with a probable error for a single measurement of less than 1 r.p.m.

Measurement of Vibrations.—It is frequently necessary to record the frequencies and amplitudes of vibrations caused by mechanical out-of-balance torsion variations and explosion impulses in high-speed engines, more especially in connection with the development of new designs. Further, it is recognized that engine

[1] U.S. Bureau of Standards, Notes, Nov. 1928. (*Franklin Inst. Journ.*)

vibration effects are undesirable from the point of view of engine supports, motor chassis and bodywork, aircraft frames, roads, etc. If, therefore, the characteristics of such vibrations can be studied from records obtained with aid of suitable apparatus, known as *Vibrographs*, the causes can be investigated and suitable modifications made in engine design.

The study of engine vibrations can very conveniently be made with the aid of the cathode ray oscillograph as explained in Chapter VIII, and both visual and photographic records obtained.

The mechanical type of vibrograph is based upon the principle of the seismograph or earthquake recorder, and it is usually arranged in two different forms or models, for vertical and horizontal vibration measurements, respectively.

It is not possible, here, to devote space to the general principles and other aspects of engine and machinery vibrations; for fuller information the reader is referred to the footnote references.[1]

Typical Vibrographs.—There have been many designs of vibrograph, the majority being employed for special research purposes, but certain commercial forms, such as the Geiger torsiograph and the Cambridge Instrument Company's instruments, are available for engine testing purposes.

The latter instruments include a vertical, horizontal, low frequency, and portable vibrograph, and also a torsional vibration damping recorder.

The vertical vibrograph (Fig. 363) is mainly intended to record small high-frequency vibrations, and is a form of seismograph comprising a weight attached to a light lever, pivoted upon knife edges and carried on a stand which is subject to the vibrations of the structure on which it is placed. The weight, by reason of its inertia, tends to remain at rest in relation to the remainder of the instrument, and causes the lever carrying the recording stylus to execute small movements corresponding to the impressed vibrations, and record them on a moving strip of celluloid 12 cm. long. The relative movements between the mass and the stand are reproduced with a mechanical magnification of ten times. For transit, the weight and therefore the moving system may be clamped. The natural period is approximately 0·25 second.

The recording drum is driven by a clockwork motor fitted with

[1] "Vibration Prevention in Engineering," A. L. Kimball (John Wiley & Sons).
"Practical Solutions of Torsional Vibration Problems," W. Ker Wilson (Chapman & Hall, Ltd.).
"Mechanical Vibrations," J. P. Den Hartog (McGraw-Hill Publishing Co. Ltd.).
"Vibration in Machinery," W. A. Tuplin, D.Sc. (Pitman, Ltd.).
"Torsional Vibration," W. A. Tuplin, D.Sc. (Chapman & Hall, Ltd.).
"Vibration Problems in Engineering," S. Timoshenko (Constable & Co., Ltd).
"The Measurement of Torsional Vibrations," R. Stansfield, *Proc. Inst. Mech. Engrs.*, July 17, 1942.

an adjustable governor enabling film speeds of from 4 to 20 mm. per second to be obtained. A time-marking mechanism operating a second stylus provides a time scale when used with a suitable time-marking accessory. The time record forms a convenient datum line from which the amplitude of the vibrations can be deduced, and may be used as a signal for marking the occurrence of events ; in addition, it enables the records of several instruments to be synchronized exactly. A suitable microscope for examining records, and other accessories are available.

The horizontal vibrograph (Fig. 364) is designed for recording small high-frequency horizontal vibrations, and employs a similar principle to that of the vertical model, but the method of suspending the mass is such that it is prevented from moving vertically, relatively to the stand ; it is free to move only in one horizontal plane. The natural period is about 0·25 second. The mechanical magnification is ten times, and a timing stylus and clamping mechanism are fitted.

The portable vibrograph (Fig. 365) is held by hand in contact at a single point with the vibrating member under test. The component of the vibration in any direction can be recorded in turn. This pattern is mainly intended to record small high-frequency vibrations, and it is particularly useful for the measurement of local vibrations in parts of machines or buildings, especially those of a recurring nature, and for analysis of resonance in parts of a structure.

The vibrations are communicated to a small rod which projects through the instrument case, the movements of which are magnified by a lever system and imparted to the recording stylus arm ; the mechanical magnification is seven times, and the mechanism is put in operation by simply depressing a push-button in the handle. In other respects the instrument is fitted with similar adjustments to the types already described. The weight is such that the pressure exerted when resting on a stationary surface causes the stylus to occupy a central zero position.

A typical record obtained with the Cambridge Portable Vibrograph is reproduced in Fig. 366.

An interesting design of vibrograph used in connection with an investigation into the vibrations of aircraft [1] and devised by the Royal Aircraft Establishment is shown, schematically, in Fig. 366. The instrument case is attached rigidly to the vibrating member and a spring-controlled mass inside the instrument remains stationary for high-frequency vibrations. The movement of the mass relatively to the case is recorded on a film driven by an electric motor ; time intervals are also recorded by a separate clock escapement device. Electromagnetic damping is employed.

[1] " Aircraft Vibration," H. Constant, Aeron. R. and M. No. 1637, Oct. 1934.

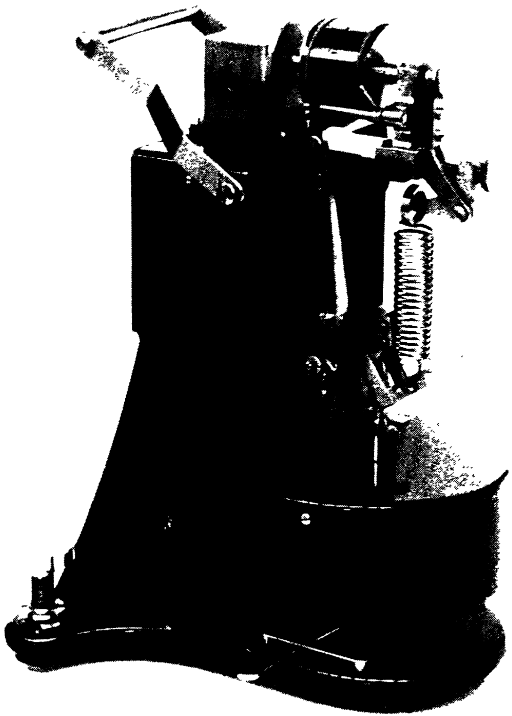

FIG. 363.—The Cambridge vertical vibrograph.

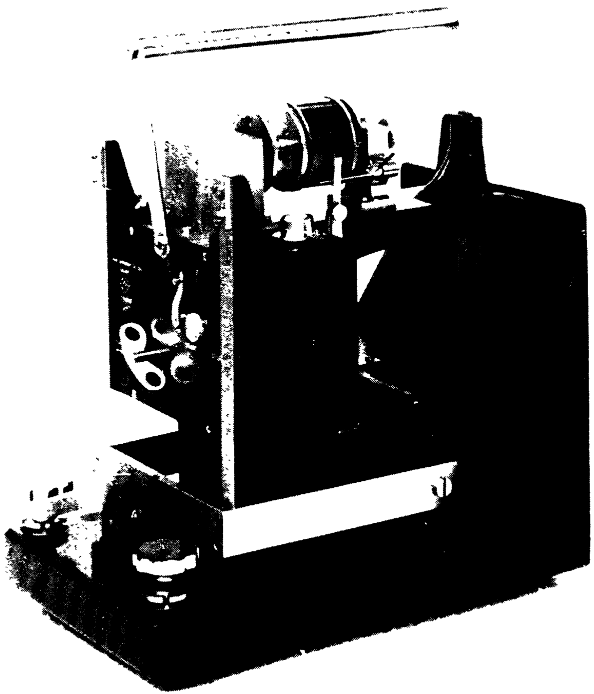

FIG. 364. The Cambridge horizontal vibrograph.

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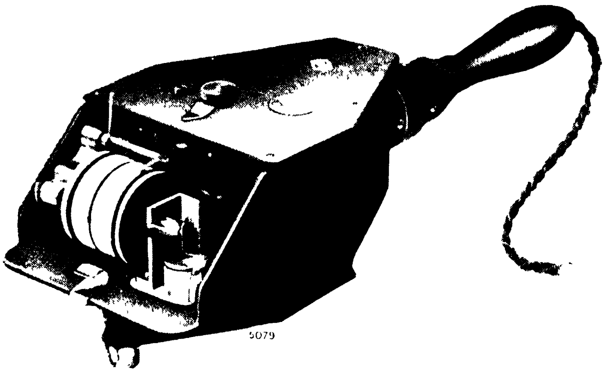

FIG. 365. Portable type of vibrograph.

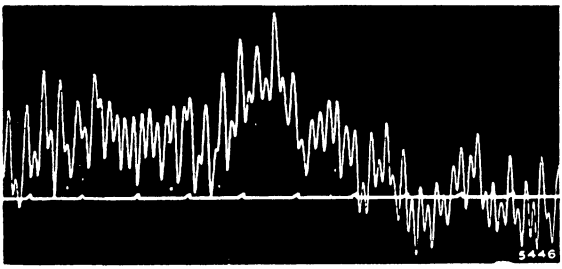

FIG. 366. Vibrograph record taken with portable instrument on cabin floor of a three-engined aircraft.

[*See page* 420.

Referring to Fig. 367 an ammeter, 3, and variable resistance, 1, are provided so that the current for the damping magnet may be adjusted to a predetermined value independent of the battery voltage. A push-button, 4, is provided on the panel for taking a record and also an independent switch, 2, to permit regulation of the damping current prior to taking a record. The vibrograph is in the form of a cube of 3¾-inch sides, weighing 2¼ lb.

There is an aluminium ring, 6, attached to the arms, 9, and moving in a cylindrical gap in the magnet, 5, energized by coils, 10. The arms are attached to the frame by means of thin steel strips, 11, which serve as control springs and as the fulcrum about which the ring and arms rotate. Recording is effected by the interposition between the 16 mm. film (not shown) and an electric lamp, 14, of

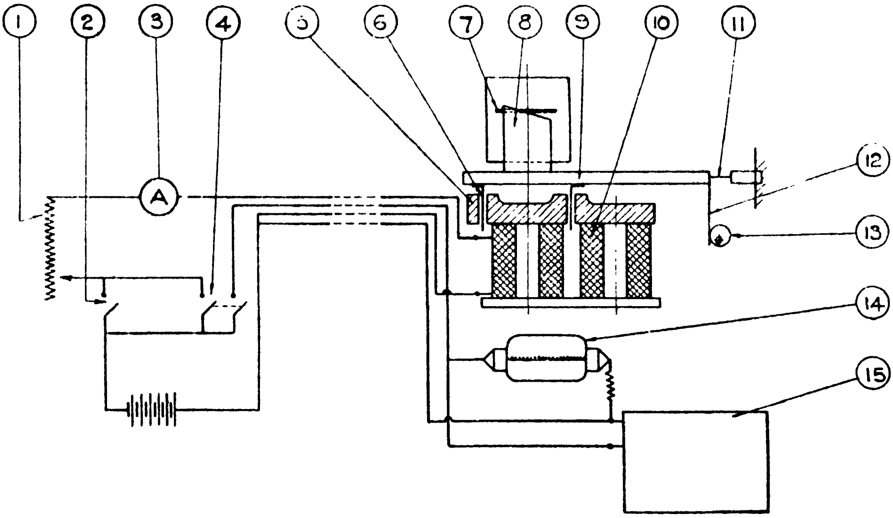

FIG. 367.—Schematic arrangement of R.A.E. vibrograph.

a narrow slit, 7, attached to the case and a thin metal tongue, 8, attached to the ring. The tongue has a straight edge inclined to the slit so that a small movement of the tongue normal to the slit causes a larger variation of the uncovered length of the slit, thus producing a magnified record of the movement of the ring relative to the case.

The film is driven by an electric motor, 15, which also drives a clock escapement (not shown) giving ten interruptions of the light per second at one edge of the film.

A subsidiary leaf spring, 12, is also provided which bears against an eccentric, 13, operated externally, so that the datum setting of the arm may be adjusted to suit different orientations of the instrument. The natural frequency of oscillation of the arm and ring on the springs is eight per second.

By using the calibration curve, the actual magnitudes of vibrations recorded having frequencies less than 1·25 times the natural frequency of the instrument may be inferred.

Using Quartz Windows for Studying Combustion Processes, etc.—Although it is possible by indirect methods to obtain a good idea of what is going on inside the cylinder of an internal combustion engine, a certain amount of speculation must be made on the results of these indirect measurements. Hitherto one has had to rely upon the indicator diagram, thermo-couples and dynamometer for analytical data on combustion phenomena.

Prior to 1914 some experiments were made by the writer in co-operation with the late Prof. W. Watson, F.R.S., on a single-cylinder petrol engine fitted with a quartz window in the combustion chamber. By using a fairly thick disc of quartz fitted in a suitable plug it was possible to observe visually, stroboscopically, and spectroscopically the actual process of combustion.

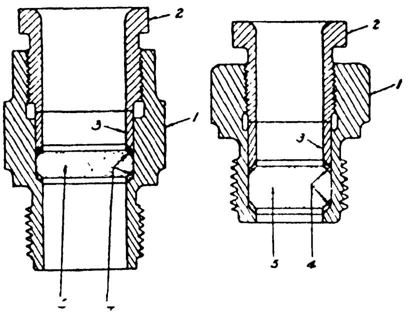

FIG. 369.—Forms of windows for combustion observations.

(1) Sparking-plug shell. (2) Gland nut. (3) Sleeve. (4) Transparent disc-sealing material ring. (5) Transparent disc.

In these experiments a rotating metal disc, having a small sector cut out in one part, was caused to rotate at one-half engine speed in front of the quartz window. By using a differential gear in the drive of this disc it was possible to make the sector uncover the quartz window at any specified part of the piston's stroke; it was thus possible to examine the nature of the combustion at any part of the piston's stroke. A number of spectrographs were taken at different parts of the engine cycle, using super-sensitized plates for the purpose. The results of these observations were eventually communicated to the Gaseous Explosions Committee of the British Association, whilst an account of the general observations is given in the writer's book, "Automobile and Aircraft Engines" (Sir Isaac Pitman).

Fig. 369 illustrates two other designs of windows for combustion observations and used by the Ethyl Corporation and National Bureau of Standards authorities. The transparent materials include quartz, spinet, sapphire, periclase and fluorite.

The transparency of these materials in the infra-red region increases in the order mentioned. The mechanical strength varies greatly, being greatest for sapphire and lowest for fluorite. The resistance to the chemical action of the combustion products varies

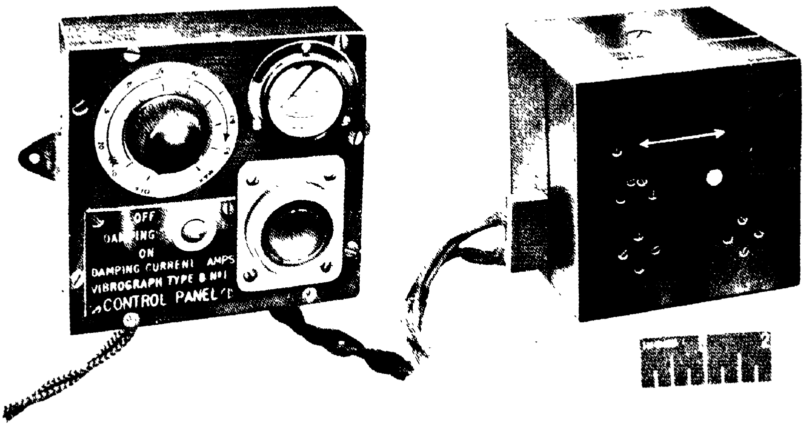

FIG. 368. R.A.E. vibrograph, type B. (*Crown copyright photograph.*)

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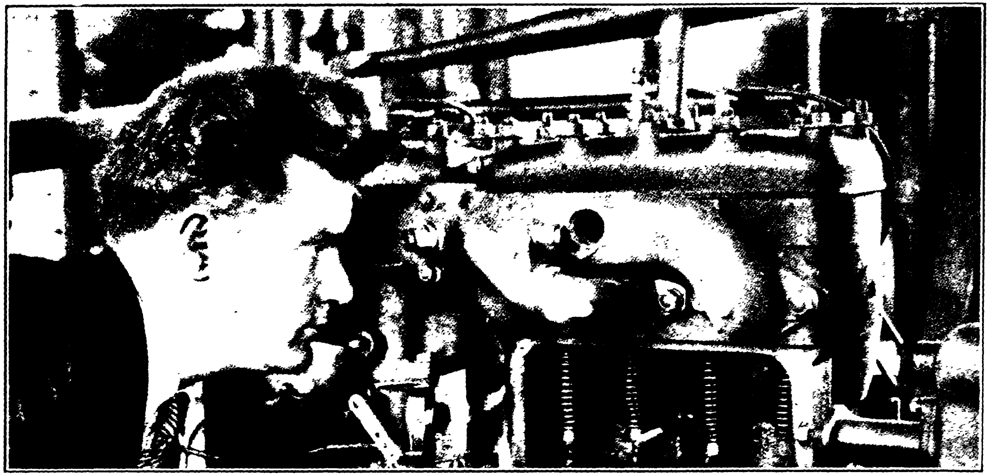

FIG. 370.—Studying inlet manifold processes with a series of quartz windows.

[*See page* 423.

widely. Sapphire is not attacked by any products of combustion, including the lead oxide which is present when leaded fuels are used. The choice of a suitable material for a given investigation will be based on a judicious balance of the four chief characteristics, namely, mechanical strength, chemical resistance, transparency, and cost.

The same method has been developed for research purposes in Germany.[1]

For the purposes of studying the processes of combustion, a single-cylinder engine has been built, the head of which is provided with numerous windows, composed of quartz glass. Something like thirty-one holes are drilled into the cylinder head. Set into these holes are small, cylindrical pieces of quartz glass, which are closed by means of plugs or perforated screws. The opening in the screws is about 3 mm., but this is ample in the case of thirty-one inspection holes to afford a survey of the processes of combustion. The luminous combustion gases reflect their light through the tiny windows on to a large optical lens which guides the rays into the observation apparatus over an inclined mirror. For the purposes of observation a stroboscope is used.

The Explosion Stroke.—The combustion process in an engine is a continuous process which is repeated constantly at very short intervals of time. Consequently, the explosions of the gas mixture which take place at lightning speed in the engine can be seen by means of the stroboscope through the quartz glass windows as a very slowly moving process. If the observation be effected through all thirty-one inspection holes, it will be seen how the spark from the sparking-plug ignites the gas mixture, how the flame spreads out and, finally, occupies the whole of the space in the combustion chamber. This is not only an interesting sight in itself, but observations carried out on these lines can result in affording very useful conclusions in regard to the speed of combustion of various gas mixtures, because it is possible accurately to measure the speed at which the explosion of the gas mixture is effected.

In these tests, the sparking plugs are connected up in such a way with neon lamps that the lamps light up as soon as an ignition spark jumps over. At the same time a line marked on the flywheel of the engine is kept under observation, which is an easy matter since in the stroboscope the flywheel appears to be standing still.

Studying the Induction and Compression Processes.—Not only can the combustion process be studied during the travel of the piston; the processes which take place during the induction of the gas mixture and on its compression during the other strokes of the piston can also be studied. For this purpose observation windows made of quartz are also arranged in the distribution

[1] *Hamburger Technische Rundschau*, July 24, 1931.

pipe of the induction piping, or in the body of the cylinder. Windows are also provided through which the interior of the cylinder and of the distribution pipe can also be illuminated by means of a powerful ray of light coming from an arc lamp. The fuel is drawn in in the form of a gaseous cloud which renders the illumination visible. Conclusions as to the composition of the gas can be drawn from the relative density of the gaseous cloud. When the gaseous cloud comes out of the distribution pipe and reaches the warmed cylinder it becomes superheated and is consequently invisible, just the same as any other superheated steam. By means of observations made through the stroboscope, it is possible to measure the period of time during which the gaseous cloud becomes invisible, so that the quantity of the moist constituents of the gas mixture can be estimated.

The consequent compression due to the upward travel of the piston results in the further heating of the gas mixture, although, on the other hand, a partial condensation sets in owing to this compression.

An Ignition Timing Indicator.—In connection with engine tests necessitating a knowledge of the effect of the ignition timing upon the performance, it becomes necessary to provide some convenient means of indicating the exact position of the spark occurrence in relation to the crank or piston of the cylinder in question.

A preliminary calibration of the ignition hand advance lever positions in relation to the crank position will enable a fairly accurate record to be kept of the point of sparking during a test.

A better method, and one that has the advantage of showing the true sparking point on the crank circle, is that employed by the Associated Equipment Company in connection with their research work.

In this method, two insulated rings (shown in Figs. 372 and 373) are employed. These are connected in series with the high-tension cable leading to one of the sparking plugs.

The inner ring is driven from the cam shaft, or magneto shaft, at one-half engine speed, although there is no reason why it should not be driven at engine speed. The outer ring is fixed.

The magneto H.T. cable is connected electrically to the inner ring, through a rubbing carbon brush. There is a small projection on the inner ring from which the spark jumps across the gap to the outer ring; the latter is connected to the sparking plug. The inner ring is timed with the engine in such a manner that the sparking point mentioned is opposite the 0° graduation on the outer fixed ring when the engine is on top dead-centre. When the engine is running and the magneto ignition advance lever is operated, the position of the spark can be read off the degrees scale on the fixed insulated ring.

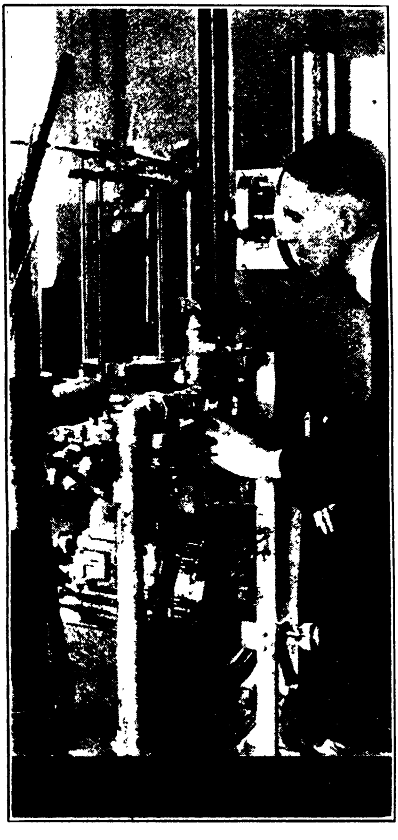

FIG. 371.—Observing combustion direct by means of a stroboscope, mirror and quartz windows.

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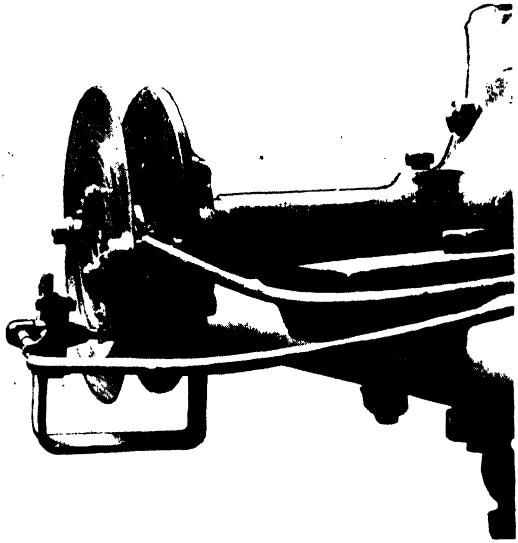

FIG. 372. The A.E.C. ignition timing position indicator.

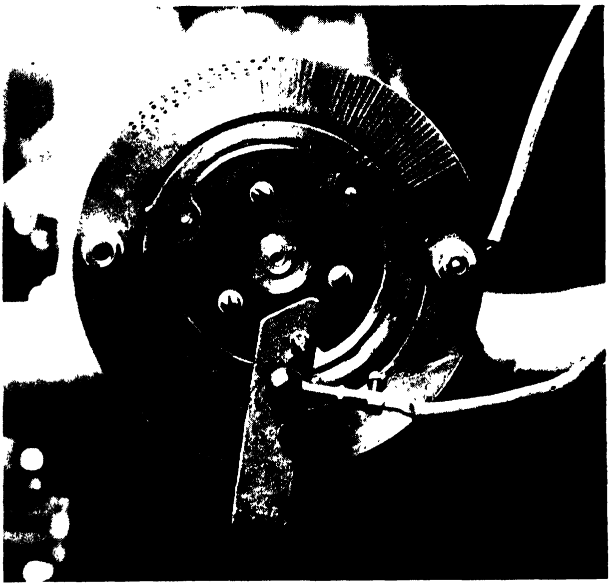

FIG. 373. Front view of the A.E.C. ignition timing position indicator.

[To face page 425.

Spark Advance Indicator for Road Tests.—Whilst it is a relatively simple matter to measure the spark advance in the case of engines in the test laboratory by means of neon tube indicators or the cathode ray indicator, it is more difficult to do this when road tests have to be made to check the operation of automatic spark advance and vacuum controlled ones as fitted to production motor vehicles.

A method devised by G. Way and S. Oldberg of the Chrysler Corporation [1] is based on the following principle: At a definite point of the cycle (60° B.T.D.C.) a relay is tripped which instantly allows current to pass through a D.C. meter. The occurrence of the spark at some later point trips another relay which stops the passage of current, and the cycle is repeated *every* engine revolution. If the meter is damped, then the average value of current as indicated by the meter is to the maximum current which flows between the events, as the interval between the events is to 360°. Thus the meter indicates the angle between these events.

Tripping at the fixed point (60° B.T.D.C.) is accomplished by an electromagnetic pick-up working from a stud in the flywheel (Fig. 374). The stud rotates past the pole ($\frac{1}{16}$-inch gap) of a permanent magnet which carries a winding, thus closing a magnetic circuit which goes through the pole, bracket, engine, flywheel and stud. A voltage is thus induced in the coil which goes slowly positive and very rapidly negative as the centre-line of the stud passes that of the pole piece, then drops to zero.

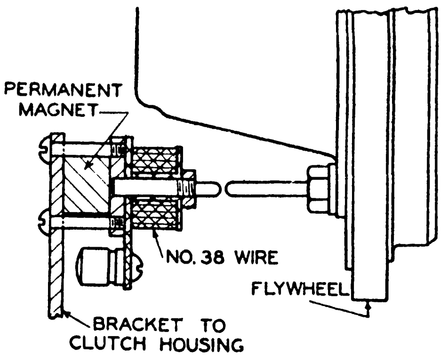

FIG. 374.—Spark advance indicator electromagnetic pick-up.

This voltage is amplified and applied to the grid of an R.C.A. 885 gas triode (see Fig. 375). This tube has the property of passing no current from plate to cathode unless the grid bias is reduced below a certain value which depends on the plate voltage, but once the current is started its value is independent of the grid voltage. The current flow is stopped by raising the cathode potential to some value positive to the plate for an instant, thus allowing the grid to gain control. In the spark advance indicator described, the amplified negative pulse generated by the flywheel pick-up reduces the bias on the gas triode valve and permits it to conduct current to the milliammeter.

[1] *Autom. Industries*, May 22, 1937.

In the spark-trip circuit, the primary of the ignition system is used to eliminate electrostatic pick-up in the internal circuits. The oscillation occurring with the spark rectified and applied to

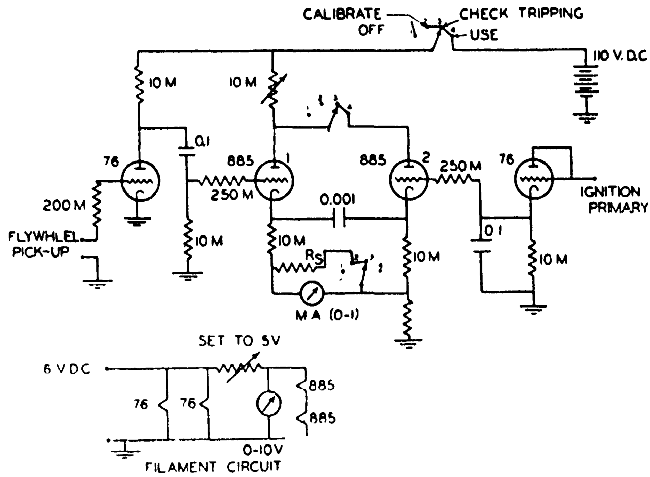

FIG. 375.—Chrysler spark advance indicator electrical and valve circuits.

the second gas triode valve, thus firing it simultaneously with the spark. This raises the cathode potential of the first valve higher than its plate, thus cutting off the current flow through the meter circuit.

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Appendix I

FORMULÆ USED IN TEST CALCULATIONS

THE relations between the various quantities which occur in engine test practice are frequently referred to in this volume. It may be useful here to summarize and to supplement the results as follows :—

(A) Fuels.—

(1) Calorific value from chemical analysis—

$$\text{C.V.} = 14{,}500\ \text{C} + 52{,}230\left(\text{H} - \frac{\text{O}}{8}\right) \text{ B.T.Us. per lb.}$$

where C, H, and O are fractional weights of carbon, hydrogen, and oxygen.

(2) Chemical equation for hydrocarbon fuel combustion with oxygen—

$$C_xH_y + \left(x + \frac{y}{4}\right)O_2 = xCO_2 + \frac{y}{2}\,H_2O.$$

(One volume of C_xH_y combines with $\left(x + \frac{y}{4}\right)$ vols. of oxygen to form x vols of CO_2 and $\frac{y}{2}$ vols. of water vapour.)

(3) 1 lb. of the fuel C_xH_y will require $\frac{32x + 8y}{12x + y}$ lb. of oxygen for complete combustion and will yield $\frac{44x}{12x + y}$ lb. of CO_2 and $\frac{9y}{12x + y}$ lb. of water vapour.

(4) 1 lb. of the fuel C_xH_y will require $4{\cdot}31 \times \frac{32x + 8y}{12x + y}$ lb. of air (= W) for complete combustion.

(5) 1 lb. of the fuel C_xH_y will require 13·14 W cu. ft. of air at 60° F.

(6) 1 cu. ft. of air-fuel mixture of calorific value (C.V.) B.T.Us. per lb. in the proportions for complete combustion, has a heating value H given by

$$H = \frac{\text{C.V.}}{13{\cdot}14\ W} \text{ B.T.Us. per cu. ft.}$$

Example.—For petrol C_6H_{14}, 15·2 lb. of air are required for complete combustion, and the equivalent volume is 200 cu. ft. The heating value of this mixture (C.V. = 19,000 B.T.Us./lb.) is 95 B.T.Us. per cu. ft., or 45 ft.-lb. per cu. in. of mixture.

(B) Efficiency—

(1) *Ideal Air Standard Efficiency* $E = 1 - \left(\frac{1}{r}\right)^{0{\cdot}404}$ where r is the compression ratio.

Engine Efficiency (Tizard and Pye).

$$E = 1 - \left(\frac{1}{r}\right)^{0\cdot296}.$$

This expression is based upon a 20 per cent. weak mixture and corresponds very closely to the conditions under which maximum observed efficiencies are now obtained.

Compression Ratios and Efficiencies (Tizard and Pye)

r	E	r	E
4·0	0·3366	6·1	0·4145
4·2	0·3461	6·2	0·4175
4·4	0·3552	6·3	0·4200
4·6	0·3637	6·4	0·4229
4·8	0·3717	6·5	0·4251
5·0	0·3790	6·6	0·4280
5·2	0·3861	6·7	0·4307
5·4	0·3930	6·8	0·4329
5·6	0·3995	6·9	0·4354
5·7	0·4026	7·0	0·4376
5·8	0·4057	7·5	0·4495
5·9	0·4087	8·0	0·4596
6·0	0·4116		

(2) *Thermal Efficiency* (*Indicated*) $= 2540 \cdot \dfrac{\text{I.H.P.}}{\text{W} \cdot (\text{C.V.})}$ where W = lb. of fuel consumed per hour, and C.V. its calorific value in B.T.Us. per lb. *Note.*—The T.E. is given as a decimal value; to convert to a percentage multiply by 100.

(3) Also *Thermal Efficiency* $= 2540 \cdot \dfrac{1}{\text{W}^1 \cdot (\text{C.V.})}$ where W^1 = weight of fuel used per I.H.P. hour.

(4) *Volumetric Efficiency* $= \dfrac{\text{V}_a}{30\ \text{C.N.}}$

where V_a = volume of air used per hour, in cu. ft. at standard temperature,
C = displacement of engine cylinders (total),
N = R.P.M.

(5) *Mechanical Efficiency* $\eta = \dfrac{\text{B.H.P.}}{\text{I.H.P.}}$

$$= \frac{\text{B.H.P.}}{\text{B.H.P.} + \text{F.H.P.}}$$

where F.H.P. = total H.P. lost in friction and pumping.

(C) Power and Torque—

(1) $\text{I.H.P.} = 9\cdot91664 \times 10^{-7} \cdot p_m \cdot d^2 \cdot l \cdot \text{N} \cdot n$

where p_m = indicated m.e.p., d = bore (ins.), l = stroke (ins.), N = R.P.M., and n = number of cylinders.

(F) Atmospheric Pressures and Temperatures, etc.—

1000 millibars = 14·496 lb. per sq. in.
= 29·5306 in. of mercury at 32° F.
1 gramme per cubic metre = 0·0006243 lb. per sq. ft.

Adiabatic Relations for Atmosphere.—

$$P \,.\, V^{1\cdot19} = \text{const.}$$

Atmospheric Density.—

$$\rho = C \,.\, P^{\frac{0\cdot19}{1\cdot19}}$$

where ρ = weight in lb. per cu. ft., P = pressure in lb. per sq. ft., C = constant (log C = $-4\cdot08836$).

Relation between temperatures and pressures—

$$T = T_0\left(\frac{P}{P_0}\right)^{1 - \frac{1}{1\cdot19}}$$

where T and T_0 are the absolute temperatures at heights corresponding to pressures P and P_0 respectively.

American Bureau of Standards formula—

$$H = 62{,}000 \log_{10} \frac{76}{P},$$

where H is the altitude in feet, and P the barometric height in centimetres of mercury.

USEFUL EQUIVALENTS, CONSTANTS, ETC.

Length

1 inch = 25·40 millimetres.
1 foot = 304·8 ,,
1 yard = 914·4 ,,
1 millimetre = 0·03937 inch.
1 mile = 1·6093 kilometres.
1 kilometre = 0·6214 mile.

Area

1 square inch = 645·16 square millimetres.
= 6·4516 square centimetres.
1 square foot = 0·0929 square metre.

Volume

1 cubic inch = 16·387 cubic centimetres.
1 cubic foot = 28·317 litres (1 litre = 1000 cubic centimetres).
1 pint = 0·5682 litre.
1 gallon = 4·5460 litres.
1 litre = 0·2200 gallon.
1 cubic centimetre = 0·0610 cubic inch.

1 gallon of water at 62° F. occupies 0·16 cubic feet (277·274 cubic inches) and weighs 10 pounds. 1 cubic foot of water at 62° F. equals 6·23 gallons and weighs 62·35 pounds.

Mass

1 grain = 0·0648 gramme.
1 gramme = 15·432 grains.
1 ounce (avoir.) = 28·350 grammes = 437½ grains.
1 gramme = 0·03527 ounce (avoir.).
1 pound (avoir.) = 453·6 grammes = 7000 grains.
1 ton = 1016 kilogrammes.

Density

1 pound per cubic foot = 0·01602 gramme per cubic centimetre.

Pressure

1 pound per square inch = 70·31 grammes per square centimetre.
1 gramme per square centimetre = 0·01422 pound per square inch.
1 ton per square inch = 1·575 kilogramme per square millimetre.

Work and Power

1 foot-pound = 1·356 joules.
1 joule = 0·7373 foot-pound.
1 English horse-power = 33,000 foot-pounds per minute.
= 1·014 metric horse-power.
= 746 watts.
1 kilogrammetre = 7·233 foot-pounds.
1 metric horse-power = 75 kilogrammetres per second.
= 0·986 English horse-power.

Heat

1 British thermal unit = 1 pound of water raised 1° F.
= 252·00 calorie.

The calorie is the heat required to raise the temperature of 1 gramme of water from $t°$ C. to $(t + 1)°$ C. The 20° calorie = $4{\cdot}180 \times 10^7$ ergs.

1 watt-minute = 14·3 calories.
The large calorie = 1000 calories.
1 B.T.U. per pound = 0·566 calorie per kilogramme.
1 B.T.U. per cubic foot = 8·9 calories per cubic metre.

Heat equivalent of 1 h.p. $= \frac{33{,}000}{778} = 42{\cdot}4$ B.T.U. per minute.

Joules' equivalent = 778 foot-pounds.
1 joule = 10^7 ergs.

Velocity

1 mile per hour = 1·467 feet per second.
= 44·70 centimetres per second.

Air

1 cubic foot of air at 14·7 lb. per square inch, and at 32° F. weighs 0·080728 pound (1·29 ounces). 1 litre weighs 1·2928 grammes.

1 atmosphere pressure = 14·7 pounds per square inch.
= 760 millimetres of mercury at 32° F. (latitude, 45°).
= $1·0132 \times 10^6$ dynes per square centimetre.
= 1·0335 kilogrammes per square centimetre.
= 29·92 inches of mercury at 32° F.
= 2116·4 pounds per square foot.
= 33·95 feet of water at 62° F.

1 pound per square inch = 2·035 inches of mercury at 32° F.
= 51·7 millimetres of mercury at 32° F.
= 2·31 feet of water at 62° F.

1 litre of air at atmospheric pressure and at 32° F. = 1·293 grammes.

1 pound of air at 62° F. = 13·141 cubic feet.

Miscellaneous

1 radian = 57·296 degrees.
π radians = 180 degrees.
π = 3·1415926.
π^2 = 9·869604.
$\sqrt{\pi}$ = 1·772453.
Value of g at London = 32·191 feet per (second)^2.
= 981·184 centimetres per (second)^2.

To convert common into hyperbolic logarithms multiply by 2·3026.
To convert hyperbolic into common logarithms multiply by 0·4343.

Density of Dry Air at Different Temperatures and Pressures (Kaye and Harker)

The densities in the table below are expressed in grammes per cubic centimetre.

Temp. °C.	Pressure in Millimetres of Mercury							
	710	720	730	740	750	760	770	780
0	·001208	·001225	·001242	·001259	·001276	·001293	·001310	·001327
2	1199	1216	1233	1250	1267	1284	1300	1317
4	1190	1207	1224	1241	1258	1274	1291	1308
6	1182	1199	1215	1232	1248	1265	1282	1298
8	1173	1190	1207	1223	1240	1256	1273	1289
10	1165	1182	1198	1214	1231	1247	1264	1280
12	1157	1173	1190	1206	1222	1238	1255	1271
14	1149	1165	1181	1197	1214	1230	1246	1262
16	1141	1157	1173	1189	1205	1221	1237	1253
18	1133	1149	1165	1181	1197	1213	1229	1245
20	1125	1141	1157	1173	1189	1205	1220	1236
22	1118	1133	1149	1165	1181	1196	1212	1228
24	1110	1126	1141	1157	1173	1188	1204	1220
26	1103	1118	1134	1149	1165	1180	1196	1211
28	1095	1111	1126	1142	1157	1173	1188	1203
30	1088	1103	1119	1134	1149	1165	1180	1195

APPENDIX II

THE CO-OPERATIVE RESEARCH METHOD FOR KNOCK DETERMINATION

THIS method has been widely used both in this country and in the United States as a means of ascertaining the knock properties of fuels. Although certain modifications have since been introduced, it still forms the principal basis of tests upon high speed petrol engine fuels for detonation characteristics.

The American Society for Testing Materials has issued details of the tentative method of test for knock characteristics of motor fuels. This method, heretofore known as the C.F.R. motor method, is intended for determining the knock characteristics, in terms of an arbitrary scale of octane numbers, of gasolines and equivalent fuels for use in spark-ignition engines, other than engines for aircraft.

The A.S.T.M. octane number of a motor fuel is the whole number nearest to the octane number of that mixture of iso-octane, with normal heptane, which the motor fuel matches in knock characteristics when compared by the procedure specified herein.

Octane number is defined by and is numerically equal to the percentage by volume of iso-octane (2, 2, 4 trimethylpentane) in a mixture of iso-octane and normal heptane, used as a primary standard for measurement of knock characteristics. Thus, by definition, normal heptane has an octane number of zero and iso-octane of 100.

Apparatus

The knock-testing unit described in this section, known as the " C.F.R engine," shall be used without modification. The apparatus shall consist of a continuously variable compression motor, together with suitable loading and accessory equipment as follows :—

(*a*) *Engine.*—Continuously variable compression, one-cylinder, with dimensions as follows : Bore, 3·25 in. ; stroke, 4·50 in. ; displacement, 37·4 cu. in. ; valve diameter, clear, 1·1875 in. ; connecting-rod bearing, diameter 2·25 in., length 1·625 in. ; front main bearing, diameter 2·25 in., length 2 in. ; rear main bearing, diameter 2·25 in., length 4·25 in. ; piston-pin, floating, diameter 1·25 in. ; connecting-rod, centre to centre, 10 in. ; timing gear face, 1 in. ; piston rings, number, 5 ; exhaust pipe, diameter 1·25 in. ; spark plug, size 18 mm. ; weight of engine (approximate), 475 lb. ; weight of complete unit (approximate), 1375 lb.

(*b*) *Crankshaft.*—Fully machined, heat-treated and counter-balanced.

(*c*) *Crankcase.*—Cast-iron, with rigid end walls.

(*d*) *Connecting-rod.*—Rifle-drilled, S.A.E. No. 1045 steel, heat-treated, bearing alloy cast directly into big-end.

(*e*) *Main Bearings.*—Renewable sleeve bushings, babbitt-lined.

(*f*) *Valves.*—Silcrome. Inlet valve with specially designed shroud.

(*g*) *Push Rods.*—Mushroom type with lock-nut adjustment.

(*h*) *Cylinders.*—Cast-iron alloy, bored and honed, Brinell hardness, 200 to 210.

(*i*) *Cylinder Head.*—Integral with cylinder.

(*j*) *Cooling System.*—Evaporatively cooled.

(*k*) *Lubrication.*—Pressure feed to main connecting-rod, piston-pin, and camshaft bearings, and to idler gear stud and gears.

(*l*) *Oil Heater.*—Electric heater in base to bring oil to operating temperature quickly.

(*m*) *Ignition.*—May be either a battery system or a magneto. A neon-tube spark indicator is built into the engine. Spark advance is automatically adjusted as the compression ratio is changed.

(*n*) *Carburettor.*—A special C.F.R. carburettor is furnished with the engine, and can be obtained with either two or four float bowls. The carburettor has a fixed fuel jet, shrouded by a variable air jet. Fuel containers are furnished with the carburettor.

(*o*) *Mixture Heater.*—As supplied by manufacturer, consisting of (1) a manifold; (2) an electric immersion heater and rheostat for controlling mixture temperature; and (3) a special mercury stem thermometer, 100 to 400° F. (38 to 204° C.), so mounted that the bulb is centred in the mixture stream to indicate the mixture temperature.

(*p*) *Instruments.*—Knock intensity is measured by a bouncing-pin, in conjunction with either a knockmeter or a gas-evolution burette. Current is supplied from a small direct-current generator, belt-driven from the power-absorbing unit. The knockmeter is a damped hot-wire ammeter which indicates the effective current in the circuit, thus permitting instantaneous readings. The gas-evolution burette is an electrolytic cell which integrates the impulses of current through the bouncing-pin circuit and indicates the total by the volume of gas collected in the burette in a given time.

(*q*) *Power-absorbing Unit.*—The engine is connected by V-type belts to an electric generator. This preferably should be an induction motor with synchronous characteristics, but may be any electric generator capable of maintaining proper operating conditions. In most cases the electric generator will act as a starting motor to crank the engine, but if a direct-current generator is used and no outside source of current is available, the engine may have to be cranked by hand.

(*r*) *Complete Unit.*—The complete unit may be obtained with the engine, generator, and panel board mounted on a cast-iron base-plate. All necessary instruments and accessories are furnished with the unit.

Reference Fuels

Primary Reference Fuels.—The primary reference fuels shall be iso-octane (2, 2, 4 trimethylpentane) and normal heptane. Both shall be certified for suitability as primary reference fuels by the United States Bureau of Standards.

Secondary Reference Fuels.—Mixtures of normal heptane and iso-octane required for referee testing are expensive. For this reason secondary reference fuels may be used for routing determinations. Such secondary reference fuels may be straight-run or other stable gasolines suitable for the purpose. One of the reference fuels should be of low knock-rating and the other of high knock-rating, or if a sufficiently high knock-rating fuel is not available, a mixture of the higher knock-rating fuel plus a knock suppressor may be used. These secondary

reference fuels shall be calibrated on the octane number scale against normal heptane and iso-octane sufficiently often to ensure accuracy of calibration ; and for every case, whether a fuel is rated by secondary reference fuels or by means of normal heptane and iso-octane, the result shall be recorded as an octane number.

For the present this certification consists of tests made by the Bureau of Standards on each batch of normal heptane or iso-octane prepared respectively by the two companies mentioned, with certificates issued to these companies authorizing them to guarantee to the purchaser that the material shipped is a part of a batch so tested and to quote the results of the Bureau of Standards tests.

Standard Operating Conditions

The engine shall be run under the following standard conditions—

(*a*) *Engine Speed.*—900 r.p.m. $\pm$ 3 r.p.m.

(*b*) *Jacket Temperature.*—Constant within $\pm$ 1° F. (0·6° C.) and at a temperature between the limits of 205° and 215° F. (96° and 102° C.).

(*c*) *Cooling Liquid.*—Distilled water, rain water, or ethylene-glycol solution when necessary at high altitude.

(*d*) *Crankcase Lubricating Oil.*—S.A.E. 30.

The viscosity range of crankcase lubricating oil, S.A.E. 30, is from 185 to 255 sec. when determined on the Saybolt Universal viscosimeter at 130° F. (54·4° C.), in accordance with the standard method of test for viscosity of petroleum products and lubricants (A.S.T.M. designation : D88) of the American Society for Testing Materials.

(*e*) *Oil Pressure.*—25 to 30 lb. per sq. in. under operating conditions.

(*f*) *Valve Clearance.*—Intake 0·008 in., cold ; exhaust 0·010 in., cold.

(*g*) *Spark Advance.*—Automatically controlled : 26° at 5 : 1 compression ratio (basic setting) ; 22° at 6 : 1 compression ratio ; 19° at 7 : 1 compression ratio.

(*h*) *Breaker-point Clearance.*—Battery system 0·015 in. ; magneto 0·020 in.

(*i*) *Spark Plug.*—Shall conform to the standard metric plug having the tolerances and thermal characteristics equal to the No. 8 spark plug furnished by the Champion Spark Plug Co., Toledo, Ohio. Gap setting, 0·025 in.

(*j*) *Throttle Opening.*—All tests shall be conducted with the throttle opening at the point of maximum volumetric efficiency, approximately 90 on the throttle scale.

(*k*) *Carburettor Adjustment.*—For maximum knock.

(*l*) *Exhaust Pipe.*—A separate exhaust pipe should be used for each engine. This pipe should be made from 1¼-in. pipe having a maximum of two ells with a total length not to exceed 20 ft. The use of a short straight-through muffler of 1¼ in. diameter passage for prevention of noise is permissible.

(*m*) *Mixture Temperature.*—The mixture temperature shall be maintained at 300° F. $\pm$ 2° F. (149° C. $\pm$ 1·1° C.) as indicated by the mercury stem thermometer.

(*n*) *Bouncing-pin Assembly.*—The gap setting shall be 0·003 in. to 0·005 in.

The following instructions for setting of the bouncing-pin contacts should be used : With the daily inspection, observe the contact points

and electrical connections to see that the points are smooth and that all the connections are tight. The gap setting should be checked (0·003 in. to 0·005 in.). The flat spring of the lower contact should touch the insulated pin with slight pressure. Too much pressure will reduce its sensitivity. To adjust the pressure accurately, set the points with 0·003 in. to 0·005 in. clearance. Then remove the diaphragm and bouncing-pin. Bend the lower spring until there is from $\frac{3}{64}$ in. to $\frac{1}{16}$ in. gap between the points. Remove the upper stop-adjusting screw and bend the upper spring until there is $\frac{5}{32}$ in. gap between the points. Check the tension on the small plunger spring in upper stop-adjusting screw and see that it has from 1 to $1\frac{1}{4}$ lb. initial tension. This can be measured by pressing it against any convenient platform scale. The pin should then be re-assembled and the adjusting screw set to give 0·003 in. to 0·005 in. gap between the points. The final adjustment is made by setting the clearance so that a knockmeter reading between 50 and 60 is obtained when operating the engine at the proper knock intensity.

Procedure

Starting and Stopping the Engine

While the engine is being turned over by electric motor, the ignition shall be turned on and the carburettor set so as to draw fuel from one float bowl. To stop the engine, the fuel and the ignition switch shall be turned off and then the motor shall be stopped by means of the push-button switch. To avoid corrosion of valves and seats between operating periods, the engine should be turned over by hand until both valves are closed.

Preliminary Adjustment of Compression Ratio

Using a mixture of 65 parts of iso-octane and 35 parts of normal heptane the compression ratio for first audible knock shall be obtained by increasing the compression ratio, by increments of two turns of the crank, from a point where there is no knock to the compression ratio at which audible knock is first detected. The proper knock intensity for use in making knock ratings shall be the knock intensity obtained with the mixture of 65 parts of iso-octane and 35 parts of normal heptane when the compression ratio is increased one unit over that compression ratio giving first audible knock. Then the numerical indication of knock intensity obtained from the knockmeter or from the gas-evolution burette, whichever is used, shall be recorded. This procedure is necessary for the first adjustment only.

This knock intensity should be equivalent to the intensity obtained with a mixture of 65 per cent. of iso-octane and 35 per cent. of normal heptane at a compression ratio of $5{\cdot}3 \pm 0{\cdot}05$ to 1 when testing at a barometric pressure of 760 mm.

For subsequent tests on fuel samples, the compression ratio shall be set to duplicate this knock intensity, as indicated by the knockmeter or the gas-evolution burette, provided no change has been made in the bouncing-pin adjustment in the meantime. In no case shall the knock intensity be such that the engine does not cease firing when ignition is interrupted.

Outline of Procedure

The octane number of a fuel shall be ascertained by comparing the knock intensity for the fuel with those for various blends of the reference fuels until two blends differing in knock-rating by not more than two octane numbers are found, one of which gives a higher knock intensity than the fuel and the other a lower knock intensity. The knock intensity shall be measured by a bouncing-pin indicator in conjunction with either a knockmeter or a gas-evolution burette.

Before the test sample and the blends of the reference fuels can be compared, the compression ratio must be set to give the proper knock intensity and the carburettor adjusted to give the maximum knock for each fuel.

Adjustment of Carburettor for Fuel under Test

Using the fuel whose knock-rating is to be determined, the carburettor shall be adjusted as follows: After one float bowl of the carburettor has been filled with the fuel of which the octane number is to be determined, adjustment shall be made to obtain maximum knock by noting the needle valve micrometer setting and the knockmeter reading or gas evolution, then turning the micrometer screw and noting whether the knockmeter reading or gas evolution increases or decreases. The micrometer shall then be turned in the direction in which the knock increases until the knock passes through a maximum. This point shall be checked three times and the micrometer set at the position of maximum knock.

(*a*) *By Knockmeter.*—When using the knockmeter, it is unnecessary to take readings over a fixed period of time, but the knockmeter needle shall be allowed to reach equilibrium after each adjustment of the micrometer.

(*b*) *By Gas-evolution Burette.*—At least two readings, agreeing within 5 per cent. of the volume of gas evolved over a period of 1 min. shall be taken as a measure of knock intensity.

Final Adjustment of Compression Ratio

Finally, adjust the compression ratio to give the same reading on the knockmeter or gas-evolution burette when using the fuel under test as was obtained in the first adjustment.

A trial blend of the low octane number reference fuel and the high octane number reference fuel, based on the expected knock-rating of the fuel sample under test, shall be placed in another carburettor float bowl and the engine run on this trial blend. The micrometer of this float bowl shall then be adjusted to the maximum knock position in a manner similar to that described.

Octane Number Determination

With the carburettor micrometers set for the air-fuel ratio of maximum knock, alternate series of readings of knock intensity shall be taken on the fuel under test and on a reference fuel blend. When using the knockmeter, the needle shall be allowed to reach equilibrium before the final reading is recorded. When using the gas-evolution burette, at least two successive 1-min. readings shall agree within 5 per cent.

At least three alternate series of readings shall be taken on each fuel. After changing from one fuel to another, at least 1 min. shall be allowed for the engine to reach equilibrium. With some fuels an appreciably longer time interval may be required. If the average knock intensity of the fuel sample is higher than the average of the reference fuel blend, the test shall be repeated with a blend containing a decreased proportion of the high octane number reference fuel. The test shall be continued in this manner until the knock intensity for the fuel sample is definitely higher than one blend and lower than another blend of the reference fuels. The difference between these two final reference fuel blends shall be not more than two octane numbers.

Calculation

The knock-rating of the fuel sample shall be obtained by interpolation from the figures so recorded, and the nearest whole number shall be reported as A.S.T.M. octane number.

Check of Test Conditions

Test conditions shall not be regarded as standard unless a blend by volume of 65 per cent. iso-octane and 35 per cent. normal heptane is matched under the specified procedure by a blend by volume of 68 $\pm$ 1 per cent. one degree benzene with 32 $\pm$ 1 per cent. normal heptane. At atmospheric pressure of 760 mm. a compression ratio of approximately 5·3 : 1 is correct for this determination. Substantial variations in atmospheric pressure will change the compression ratio required.

For routine checking purposes, one degree benzene and secondary reference fuels may be used.

" One degree " benzene is a commercial product conforming to the following requirements :—

(*a*) Boiling range not greater than 1° C., embracing the boiling-point of chemically pure benzene.

(*b*) Specific gravity of from 0·882 to 0·886, when determined in accordance with the tentative method of test for gravity of petroleum and petroleum products by means of the hydrometer (D287-32T) of the American Society for Testing Materials.

(*c*) Free from hydrogen sulphide, carbon disulphide, and thiophene.

(*d*) Free from turbidity, with colour not darker than a solution of 3 mg. of potassium dichromate in 1 litre of water when compared in 50 c.c. Nessler tubes.

Accuracy

Results obtained by this procedure with different C.F.R. engines and in different laboratories should differ by not more than two octane numbers.

While the accuracy of this method is as above stated, its sensitivity may permit detection of differences as small as 0·2 octane number when alternate readings of knock intensity are taken on the two fuels as directed.

The reader should also consult the recent Air Ministry Engine Testing and Fuel Specifications for particulars of modifications to the C.F.R. method of testing fuels.

Appendix III

INSTITUTION OF AUTOMOBILE ENGINEERS' LABORATORIES

The research laboratories of The Institution of Automobile Engineers, Great West Road, Brentford, Middlesex, are equipped for carrying out a very large number of tests on engines, complete vehicles and chassis components as well as for the undertaking of special research work. The work of the committee is financed partly by grants from the Government Department of Scientific and Industrial Research and from the Society of Motor Manufacturers and Traders. In addition, individual companies wishing to take part in this co-operative effort become affiliated to the Institution and contribute towards the cost of the work. Many manufacturers and operators contribute annual subscriptions. The programme of research (1938-39) included the following items :—

Cylinder, piston and piston ring wear ; brake squeak ; wear of bearing metals and crankshafts ; lubricating oil consumption ; cold starting of petrol engines ; gear wear and durability ; impact of tyres on roads ; properties of materials affecting deep-drawing performance ; oxidation of lubricating oils ; engine friction at high speeds ; measurements of engine bearing temperatures ; silencing of motor cycles ; heat transfer from brake drums to wheel rims ; fatigue failures of crankshafts ; mixture distribution ; effect of radio interference suppressors on engine performance.

Previous to this period over 75 research reports had been issued and in addition a total of 3350 abstracts prepared from British and foreign publications and circulated in loose-leaf form to affiliated members.

In general the principal aim of the committee is to encourage co-operative research on problems of common interest to the motor industry. Research of this kind results in a general improvement and development in vehicles, in increased popular demand, and, therefore, in a more prosperous industry. It is, therefore, to the interest of all manufacturers and operators associated directly or indirectly with the motor industry, to support the work of the committee.

The experimental work is undertaken either at the Institution's laboratory or at any other suitable research centre, the programme of research being determined following an exhaustive enquiry amongst manufacturers and operators.

Manufacturers and operators can themselves co-operate in the work of the committee by contributing their own experience on any particular problem, and such co-operation has already proved very effective.

Appendix IV

SUPERCHARGER CALCULATIONS

The following empirical formulæ are used in connection with the Air Ministry's official type tests on superchargers :—

W = mass of air flowing per minute (in. lb.).
C_P = specific heat at constant pressure (0·238).
γ = ratio of specific heats $\frac{C_P}{C_\gamma}$ (1·41).
T_1 = absolute temperature of air intake (°C.).
T_2 = absolute temperature of air delivery (°C).
P_1 = absolute pressure of air intake (lb. per sq. in.).
P_2 = absolute pressure of air delivery (lb. per sq. in.).
J = Joule's equivalent (1400 ft.-lb. per C.H.U.).

(1) Horse-Power for Adiabatic Compression—

$$\text{Adiabatic h.p.} = \frac{WJC_PT_1}{33{,}000}\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right].$$

(2) Actual Compression Horse-Power—

$$\text{Actual h.p. input to air} = \frac{WJC_P}{33{,}000}(T_2 - T_1).$$

(3) Overall Adiabatic Efficiency—

$$\text{Adiabatic efficiency} = \frac{WJC_PT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{33{,}000 \times \text{observed h.p. input}}.$$

Note.—The adiabatic efficiency of an air compressor is the ratio of the power that would be required, theoretically, to compress the air adiabatically from the lower to the higher pressure, to the measured power input, assuming no mechanical losses occur.

(4) Volumetric Efficiency—

$$\text{Volumetric efficiency} = \frac{1{\cdot}355\, WT_1}{DNP_1},$$

where D = theoretical displacement of rotor (cu. ft. per revolution),
N = r.p.m. of rotor.

(5) Adiabatic Temperature Efficiency.—This is defined as the ratio of the adiabatic h.p. to the actual h.p. input to the air.

$$\text{Adiabatic temperature efficiency} = \frac{T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_2 - T_1}.$$

(6) Power to Drive Blower.—The approximate power absorbed by a supercharger is given by the following relation, which assumes an overall efficiency of 0·58 for the blower :—

Horse-power absorbed by blower =

$$\text{engine B.H.P.}\left[\frac{0\cdot00208\,T_1(r^{0\cdot291}-1)}{1-0\cdot00208\,T_1\,(r^{0\cdot291}-1)}\right]$$

where T_1 = intake temperature and r = compression ratio of blower, i.e. final to initial (absolute) pressures.

(7) Corrected Compression Ratio.—As the intake temperature at the rated altitude is less than the ground intake temperature, when the supercharger is tested at the reduced atmospheric pressure corresponding to the rated altitude, it is necessary to correct the ground level compression ratio in order to allow for this effect. The following empirical formula enables this correction to be made :—

$$\text{Corrected ratio } r = r_0[1 + 0\cdot00063 r_0^2(T_1 - T_a)]$$

where r_0 is the observed compression ratio $\left(\frac{P_2}{P_1}\right)$ and T_a is the absolute temperature, °C., at the rated altitude.

(8) Corrections for Pressure and Temperature.—When a supercharged engine is tested for power output on the ground, or sea-level conditions, the measured b.h.p. requires a correction for the difference between the charge and exhaust back pressure at any altitude. The following empirical formula is employed for this purpose :—

Corrected b.h.p. =

$$\text{Test b.h.p.}\left[1 + \frac{760 - \text{barom. press. at given altitude}}{3500}\right].$$

Note.—The barometric pressure must be expressed in terms of millimetres of mercury.

There is also a correction for temperature to the sea-level b.h.p., namely, as follows :—

$$\text{Corrected b.h.p.} = \text{test b.h.p.}\sqrt{\frac{T_1}{T_a}}$$

where T_1 and T_a are the absolute temperatures °C. given in (7).

Appendix V

THE CALCULATION OF THE QUANTITY OF AIR WHICH PASSES THROUGH A THROTTLE PLATE

For convenience in rapidly calculating the quantity of air which has passed through a given orifice under given conditions, the numbers tabulated in the Tables below have been calculated. The volume of air in cubic centimetres and cubic feet, and the weight of air either in grammes or pounds, which would pass through an orifice of given diameter under a pressure difference of 1 inch, if the barometric height were 760 mm., the temperature of the air 15° C. (59° F.), and the hygrometric state one of half saturation, are given in the Sub-Table A. The numbers in Sub-Table A are obtained by using the values of α deduced with the large box when the pressure difference was steady. If the pressure difference is variable, the value of a, that is, the amplitude of the variation divided by the mean pressure, must be obtained, and then the value of the correcting factor c should be taken from Fig. 91 (p. 133).

If the conditions vary from the above standard values, correcting factors are to be used to reduce the quantities given in Sub-Table A. Thus in the Sub-Table B are given factors which correct for the amount that the pressure difference varies from the standard value taken as 1 inch. Factors in Sub-Table C correct either the volume or the weight for temperatures differing from 15° C., and those in Sub-Table D make the necessary correction for the barometric height differing from 760 millimetres.

In Sub-Table E there is a factor to allow for the change in the coefficient α which is caused by the fact that the pressure difference is not 1 inch.

SUB-TABLE A

Tables for Calculating the Quantity of Air Flowing through a Throttle-Plate

Calculated for :—
Time of flow = 1 second.
Temperature = 15° C. (59° F.).
Barometer = 760 mm.
Hygrometric state = half saturated at 15° C.
Pressure difference of 1 in. of water.

Diameter of Orifice In.	Coefficient α	Volume of Air Passing through Orifice (Half Saturated). Cm³.	Volume of Air Passing through Orifice (Half Saturated). Cubic Feet	Weight of Air (Half Saturated). Grams	Weight of Dry Air. Grams	Weight of Dry Air. Lb.
$\frac{3}{8}$	0·6096	877·3	0·03066	1·072	1·066	0·00235
$\frac{1}{2}$	0·6073	1553	0·05487	1·899	1·889	0·00416
$\frac{5}{8}$	0·6041	2414	0·08527	2·950	2·935	0·00647
$\frac{3}{4}$	0·6027	3469	0·1226	4·239	4·217	0·00930
$\frac{7}{8}$	0·6010	4708	0·1663	5·753	5·723	0·0126
1	0·5997	6135	0·2168	7·498	7·459	0·0164
$1\frac{1}{8}$	0·5985	7749	0·2737	9·469	9·420	0·0208
$1\frac{1}{4}$	0·5978	9557	0·3376	11·68	11·62	0·0256
$1\frac{3}{8}$	0·5971	11550	0·4081	14·14	14·07	0·0317
$1\frac{1}{2}$	0·5966	13730	0·4851	16·78	16·69	0·0368
$1\frac{5}{8}$	0·5961	16100	0·5687	19·67	19·57	0·0431
$1\frac{3}{4}$	0·5957	18660	0·6592	22·80	22·68	0·0500
$1\frac{7}{8}$	0·5954	21410	0·7563	26·17	26·03	0·0574
2	0·5951	24350	0·8602	29·77	29·62	0·0653
$2\frac{1}{8}$	0·5948	27480	0·9712	33·57	33·40	0·0736

SUB-TABLE B

	Correction for Pressure Difference			Correction for Pressure Difference	
At Pressure Difference. Inches of Water	Multiply Weight and Volume by	Difference	At Pressure Difference. Inches of Water	Multiply Weight and Volume by	Difference
0·50	0·707	0·068	1·30	1·140	0·046
0·60	0·775	0·062	1·40	1·183	0·043
0·70	0·837	0·057	1·50	1·225	0·042
0·80	0·894	0·055	1·60	1·265	0·040
0·90	0·949	0·051	1·70	1·304	0·039
1·00	1·000	0·049	1·80	1·340	0·036
1·10	1·049	0·045	1·90	1·378	0·038
1·20	1·094		2·00	1·415	0·037

SUB-TABLE C

Correction for Temperature

At Temperature °C.	Multiply Volume by	Multiply Weight by	At Temperature °C.	Multiply Volume by	Multiply Weight by
0	0·973	1·026	16	1·001	0·998
1	0·975	1·024	17	1·003	0·996
2	0·977	1·023	18	1·005	0·994
3	0·979	1·021	19	1·007	0·992
4	0·980	1·019	20	1·009	0·991
5	0·982	1·017	21	1·010	0·989
6	0·984	1·016	22	1·012	0·987
7	0·986	1·014	23	1·014	0·986
8	0·987	1·012	24	1·016	0·984
9	0·989	1·010	25	1·017	0·982
10	0·991	1·009	26	1·019	0·980
11	0·992	1·007	27	1·021	0·979
12	0·994	1·005	28	1·023	0·977
13	0·996	1·003	29	1·024	0·975
14	0·998	1·001	30	1·026	0·973
15	1·000	1·000			

SUB-TABLE D

Correction for Barometer

At Barometric Height. Millimetres	Multiply Volume by	Multiply Weight by	At Barometric Height. Millimetres	Multiply Volume by	Multiply Weight by
730	1·021	0·980	756	1·003	0·997
732	1·019	0·981	758	1·002	0·999
734	1·018	0·982	760	1·000	1·000
736	1·017	0·983	762	0·999	1·002
738	1·015	0·985	764	0·997	1·003
740	1·014	0·987	766	0·996	1·004
742	1·013	0·988	768	0·995	1·006
744	1·011	0·990	770	0·993	1·007
746	1·010	0·991	772	0·992	1·008
748	1·008	0·992	774	0·991	1·010
750	1·007	0·993	776	0·990	1·011
752	1·006	0·995	778	0·988	1·013
754	1·004	0·996	780	0·987	1·014

SUB-TABLE E

Correction for Variation of α with Pressure Difference

At Pressure Difference. Inches of Water	Multiply Weight and Volume by	At Pressure Difference. Inches of Water	Multiply Weight and Volume by
0·50	1·003	1·30	0·999
0·60	1·002	1·40	0·998
0·70	1·002	1·50	0·998
0·80	1·001	1·60	0·997
0·90	1·001	1·70	0·997
1·00	1·000	1·80	0·996
1·10	0·999	1·90	0·996
1·20	0·999	2·00	0·995

APPENDIX VI

REFERENCE has been made (pp. 160, 180) to the use of dynamometers of the Eddy Current type. This form of dynamometer is now coming into considerable prominence, both in this country and in America, where a recent design has been very successful. This machine is known as the Dynamatic Dynamometer, for which the licence in all countries except the North American continent is held by Heenan & Froude, Worcester.

This dynamometer can be made in sizes capable of absorbing very large powers, such as those of big marine diesel engines, and is extensively

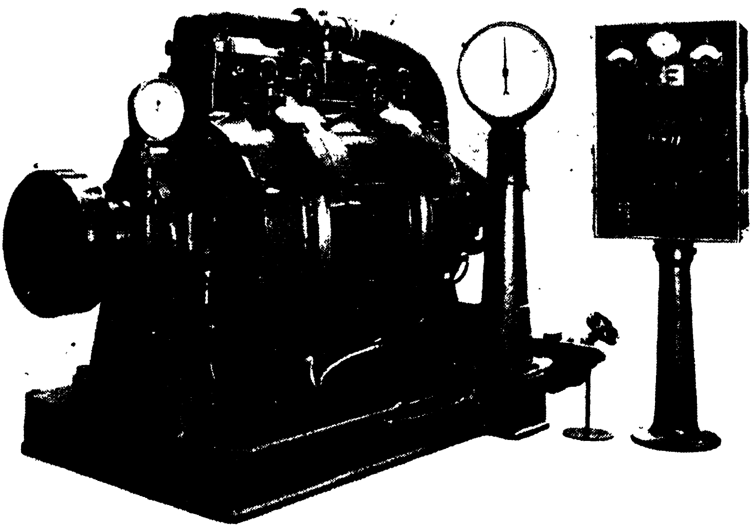

FIG. 1.—Dynamatic Dynamometer capable of absorbing 3,000 B.H.P. (*Courtesy of Messrs. Heenan & Froude Ltd*).

employed for testing aircraft engines. It consists in effect of a steel rotor, without any electrical coils, windings, slip-rings, etc., rotating in the magnetic field of a surrounding stationary coil; the resistance to rotation at any given speed depends upon the extent to which the coil is excited, and as the maximum excitation current on even a very large dynamometer only amounts to a few kw., light and sensitive control gear can be employed. Owing to the special profile of the rotor, which resembles that of a toothed gear, electrical characteristics are obtained which give very stable operation.

Fig. 1 is an illustration of a Dynamatic Dynamometer supplied to a North of England builder of land and marine diesel engines, capable of absorbing 3,000 B.H.P. For testing such engines a Dynamatic Dynamometer has the useful feature that it is inherently reversible, absorbing

power equally well in either direction of rotation ; the control panel can, of course, be set down in whatever position is most convenient to the tester.

A twin unit, arranged for testing aircraft engines up to 4,000 B.H.P., is shown in Fig. 2. The layout is such that both units can rotate in the same directions for testing normal engines, or can rotate in opposite directions for testing engines fitted with contra-rotating shafts. The small motor and reduction gearbox on the bedplate extension are for motoring and starting the engine.

The essential elements of this dynamometer (the rotor and the surrounding coil) are used to form the basis of a new form of Heenan Regenerative Dynamometer which converts a portion of the engine power into useful electric current. By mounting the coil (or coils) in a member which can rotate in its own bearings, the device becomes a magnetic eddy-

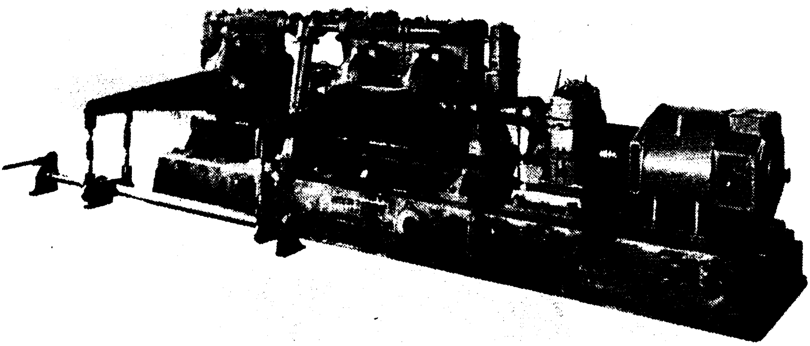

FIG. 2.—Twin Unit Dynamatic Dynamometer for Testing Aircraft. Engines up to 4,000 B.H.P.
(*Courtesy of Messrs. Heenan & Froude Ltd.*)

current coupling which can transmit the entire engine torque while "slipping" at a controlled rate. In this dynamometer, the engine drives one member of the coupling, while the other member drives an alternator ; the latter is connected to the works A.C. network which, being at grid frequency, maintains the alternator speed constant, and the engine speed is regulated by varying the excitation of the coupling and thus its slip. The alternator can, of course, also act as a motor, to rotate the engine at any desired speed *via* the coupling.

For the few occasions on which tests of engine power output are required at speeds below the synchronous speed of the alternator, the latter is automatically taken off the line and brought to rest, where it is locked by a friction brake. In these conditions the coupling, with one member stationary, acts as a Dynamatic Dynamometer which can, in fact, absorb the full engine power continuously. The machine thus allows testing to proceed without interruption, even at times when it may

not be desirable to regenerate the engine power, a most important feature.

An illustration of a typical Regenerative Dynamometer of this type is given in Fig. 3, this referring to a machine for testing aircraft engines up to 4,000 B.H.P.

For such work it is now the practice to control the excitation of the coupling, and thus the load upon the engine, by electronic means. This control gear is usually designed to give a " cube-law " characteristic, i.e. the power absorbed with the controls unaltered varies as the cube of the speed, just as with a fixed-pitch airscrew or marine propeller. At the same time there is an adjustable control which overrides the " cube-law " at any pre-determined speed and which provides a very much sharper increase of power with any further increase in speed ; this resembles the characteristics of a variable-pitch airscrew, when in flight under the control of a constant-speed mechanism.

Over 100 Dynamatic Absorption or Regenerative Dynamometers are

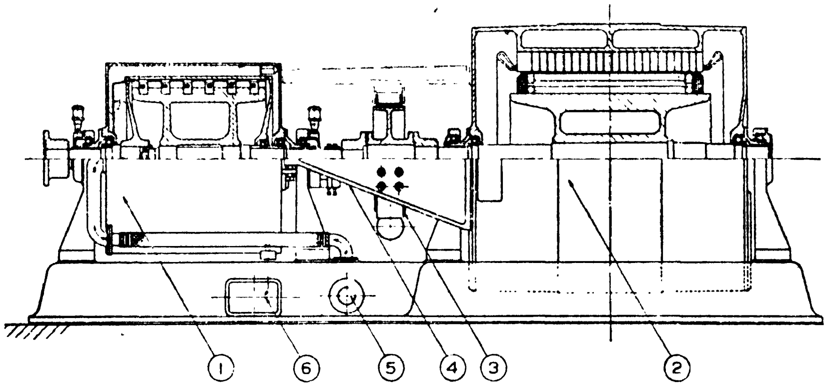

FIG. 3.—Diagrammatic Layout of Heenan A.C. Regenerative Dynamometer.
Key. (1) Dynamatic coupling brake. (2) Alternator. (3) Friction brake. (4) Torque yoke. (5) Water inlet. (6) Water outlet.

in use in America on aircraft engine testing alone, most of them having been operating at this date for thousands of hours. Considerable numbers have also been supplied to builders of aircraft engines, diesel engines, etc., in this country, many more being under construction.

In connection with the actual energy recovery, some quantitative tests * made by Pratt and Whitney Aircraft Co. of American upon an aircraft engine connected through an hydraulic coupling to an A.C. generator, showed that of the maximum continuous output of 1,550 B.H.P. generated by the engine, 930 H.P. was made available for useful purposes by the generator. It was demonstrated that the power lost was proportional to the degree to which the engine operated above the synchronous speed of the generator.

Fig. 4 has been prepared from test data, and shows in greater detail the character of the power distribution. Curve A shows the engine output for a typical load test. Curve C denotes the power returned to

* "The Power-Recovery System of Testing Aircraft Engines," G. E. Cassidy, W. L. Wright, and W. A. Mosteller., *Aero Digest*, February, 1943.

the power system through the generator. The difference between the ordinates of curves A and C is the total power lost and is made up of the true slip loss A—B, as mentioned previously, and the transmission loss B—C. The latter loss is made up of two parts, namely, the losses in the slip-coupling and those in the loading generator. Tests show that the transmission losses are fairly constant for all loading conditions, and amount to approximately 5 per cent. of the engine output. It is under

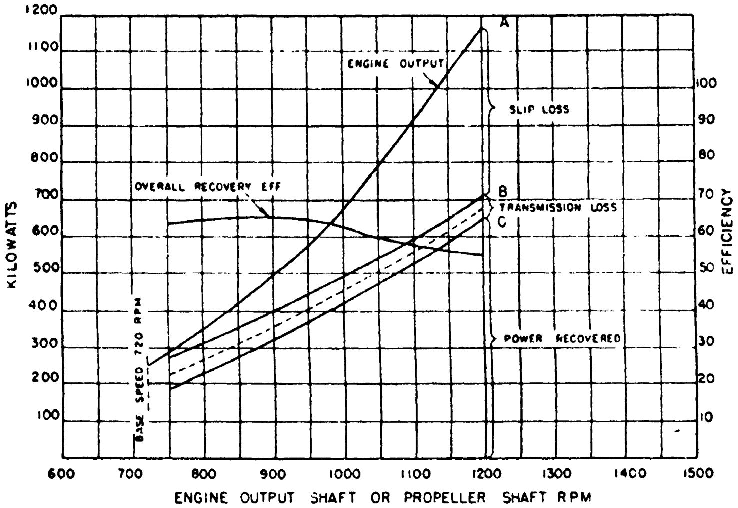

FIG. 4.—Power Utilization and Efficiency Curves (Pratt and Whitney).

stood that the English Electric Company, Stafford, has evolved a satisfactory arrangement of power recuperation, comprising a swinging frame D.C. electric dynamometer electrically connected to a D.C. motor which is coupled to an A.C. generator on the Ward-Lennox system. Manual or automatic control of the dynamometer and motor fields is used to enable a suitable electrical supply to supplement the existing work's supply to be provided. It is stated that over 80 per cent. of the engine output is recoverable as useful energy.

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